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EVALUATION OF PLASMA JET IGNITION FOR IMPROVED PERFORMANCE OF A--ETC(U)

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# EVALUATION OF PLASMA JET IGNITION FOR IMPROVED PERFORMANCE OF ALTERNATE FUELS

FINAL REPORT
AUGUST 1982

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U.S. ARMY MOBILITY EQUIPMENT RESEARCH
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NEWTON UPPER FALLS, MA 02164

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Alcohols, such as ethanol and methanol, are potential substitutes for gasoline during periods of fuel shortages. The pure alcohols have been reported to cause performance and starting problems when used to fuel internal combustion engines. This study characterized how three modes of ignition, OEM-magneto, high energy conventional spark (CI) and plasma jet ignition (PJI) influenced the engine combustion properties of ethanol, methanol and gasoline-alcohol blends. Specific combustion properties examined in these measure-

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ments were burning velocity and lean limit. In addition, the engine performance was determined for 30% alcohol-gasoline containing blends. These engine performance measurements determined brake power, brake specific fuel consumption and brake emissions of carbon monoxide and hydrocarbons. The findings of this study suggest that high energy ignition systems, such as plasma jet ignition, will improve both fuel combustion properties and engine performance.

TABLE OF CONTENTS

## LIST OF FIGURES

## LIST OF TABLES



## LIST OF SYMBOLS AND DEFINITIONS

| | |
|---|---|
| $\phi$ | Equivalence ratio, actual fuel/air ratio divided by the stoichiometric fuel/air ratio. |
| AF | Mass air divided by mass fuel |
| BTDC | Before top dead center |
| BP | Brake specific power |
| BSFC | Brake specific fuel consumption |
| BSHC | Brake specific hydrocarbons |
| CI | Convention spark plug and the PJI high energy power supply. |
| PJ, PJI | Plasma jet, Plasma jet ignition |
| TDC | Top dead center |

# EXECUTIVE SUMMARY

Since the supply of Middle Eastern oil is uncertain and our military must be battle-ready under any situation, alternate fuels are being considered to power spark ignited engines in vehicles, generator sets and other small engine powered equipment. Alcohols, such as ethanol and methanol, appear to be viable alternatives to gasoline. This study investigates how the mode of ignition influences the combustion and engine performance characteristics of the alcohols and alcohol-gasoline blends. The findings of this study suggest that high energy ignition systems, such as plasma jet ignition, will improve both fuel combustion properties and engine performance.

Plasma Jet Ignition (PJI) has been found to be a superior ignition concept. PJI was found to produce enhanced burning velocities in both one atmosphere combustion chamber experiments and optical engine measurements. The PJI enhanced burning velocity was demonstrated for all fuel types. When compared to a high energy conventional spark ignition (CI), PJI extended the lean limit of the ignition of mixtures in the optical engine by more than a factor of 10%. A hydride fueled PJI plug was found to produce significantly enhanced burning rates in relation to either the unfueled PJI or the CI ignitors. The superior results using the fueled PJI suggested that an engine compatible plug be developed and tested. (These tests are necessry to determine if the enhanced performance of the fueled PJI will translate

into improved engine performance.)

This study also determined that the alcohol-gasoline blends (<30%) did not dramatically reduce the typical internal combustion performance or fuel consumption compared to that obtained using gasoline. However, the 30% methanol-gasoline blend was found to produce significantly increased aldehyde exhaust emission levels as compared to those determined for 30% ethanol-gasoline and neat gasoline. These results indicated that ethanol was the optimum gasoline substitute in terms of engine performance, exhaust emissions and reduced cold starting problems. Substitution of 30% ethanol for gasoline in many tests produced improved engine and combustion properties.

This study recommends that an improved PJI plug be constructed and tested. Multi-cylinder engine performance evaluation of the high energy ignition systems is suggested to further characterize engine performance, response to load, fuel compatibility, fuel consumption and low temperature (0°C) starting characteristics. More extensive engine testing of alcohol-gasoline blends is recommended to determine the maximum percentage of alcohol substitution which will provide acceptable engine performance. This study further recommends that small engines, such as those used in power generator sets, which employ carburetion for the fuel delivery should be modified to have fuel injection. Fuel injection will provide greater fuel type flexibility and faster fuel conversion.

# 1.0 INTRODUCTION

## 1.1 Background

The current generation of spark-ignited internal combustion (SI) engines resulted from a simultaneous evolution of both engine design and fuel refining. During this evolutionary period the engine designer was relatively free to specify the fuel properties necessary to meet engine performance requirements. In the past, the availability of highly refined fuel was unquestioned. This view (belief) was radically changed by the sharp increase in the price of crude oil and the 1973 oil embargo.

Since the SI engine represents a significant portion of the engine types used for military transportation and power generation, various materials, such as ethanol and methanol, have been suggested as possible alternate fuels in times of gasoline shortages. Alcohols have been historically used in times of fuel shortages, i.e., Germany in World War II. Presently, Brazil is powering its vehicles on a mixture of ethanol and gasoline.[1] Further, Brazil intends to rely entirely on ethanol as automotive fuel. Gasohol is currently offered by service stations in the United States. Considering the reserves of coal and biomass in this country from which alcohol can be produced, the substitution of alcohol for gasoline as an alternate to gasoline as fuel for SI engines would be advantageous.

## 1.2 Alcohol as Fuel

Methanol and ethanol, either neat or as alcohol-gasoline blends are considered as viable alternate fuels. Alcohol containing fuels have some attractive characteristics as

automotive and small engine fuels. The combustion of alcohol containing fuels in the SI engine results in improved knock resistance, increased indicated mean effective pressure, extended combustion limits and reduced exhaust emissions. However, since the performance of the SI engine has been optimized for burning gasoline, it will not be optimal for burning alcohol containing fuels. Primary considerations raised about the performance of alcohol fuels center on the different physical and thermodynamic properties associated with alcohols in relation to gasoline. Table 1 compares these properties with gasoline. A comprehensive review of studies related to alcohols as automotive fuels can be found elsewhere.[2,3] The applicable data to SI engines is discusse below.

### 1.2.1 Neat Methanol

Methanol can be produced from various materials such as coal, natural gas, natural waste products and wood. Since methanol is available in industrial quantities, it has been extensively studied in laboratory engines and cars. Methanol burns with greater efficiency and more power when compared to gasoline. Efficiency improvements obtained with methanol are due to methanol's lower combustion temperature and higher flame speed. In addition, methanol combustion generates more moles of product gas per mole of air as compared to gasoline combustion.[4] Improved power output of the methanol fueled engine results from the higher energy continent of the fuel/air charge. However, this improved engine performance is accompanied by an increase in the specific fuel consumption.

## Table 1

## FUEL PROPERTIES

| CHARACTERISTIC | GASOLINE | NO 1 DIESEL FUEL | NO 2 DIESEL FUEL | ETHANOL | METHANOL | GASOHOL |
|---|---|---|---|---|---|---|
| CHEMISTRY | MIXTURE OF HYDROCARBONS | MIXTURE OF HYDROCARBONS | MIXTURE OF HYDROCARBONS | $C_2H_5OH$ | $CH_3OH$ | 90% UNLEADED GASOLINE 10% ETHANOL |
| Approx Specific Gravity @ 60°F | 72 - 75 | 82 | 85 | 79 | 79 | 73 - 76 |
| Boiling Point | | | | | | |
| °F | 86 - 437 | 360 - 530 | 375 - 630 | 173 | 149 | 77 - 410 |
| °C | 30 - 225 | 190 - 280 | 210 - 325 | 78 3 | 65 | 25 - 210 |
| Net Heating Value (Mass) | | | | | | |
| BTU/lb | 18.700 | 18.500 | 18.400 | 11.600 | 8.600 | 18.000 |
| MJ/kg | 43 5 | 43 | 43 | 27 | 20 1 | 41 9 |
| Net Heating Value (Volume) | | | | | | |
| BTU/gal | 117.000 | 126.000 | 130.000 | 76.000 | 57.000 | 112 900 |
| MJ/1 | 32 | 35 3 | 36 6 | 21 3 | 15 9 | 30 9 |
| Heat of Vaporization | 170 | 250 | 250 | 390 | 500 | 200 |
| kJ/kg | 400 | 600 | 600 | 900 | 1.110 | 465 |
| Vapor Pressure @ 100°F | | | | | | |
| psi | 9 - 13 | 05 | .04 | 2 5 | 4 6 | 8 - 16 |
| kPa | 62 - 90 | .34 | .27 | 17 | 32 | 55 - 110 |
| Octane Number | | | | | | |
| Research | 91 - 100 | Note 1 | Note 1 | 111 | 112 | Note 2 |
| Motor | 82 - 92 | | | 92 | 91 | |
| Cetane Number | Below 15 | 40 - 60 | 40 - 60 | Below 15 | Below 15 | Below 15 |
| Stoichiometric A/F Ratio | 14 6 | 14 6 | 14 6 | 9 | 6 4 | 14 |
| Vapor Flammability Limits, % by Volume | .6 - 8 | .6 - 6 5 | 6 - 6 5 | 3 5 - 15 | 5 5 - 26 | Note 3 |
| Viscosity @ 40°C | | | | | | |
| Centipoise | .5 | 1 45 | 2 41 | 83 | 46 | 5 |
| Centistokes | .6 | 1 75 | 2 79 | 1 1 | 58 | 6 |
| Appearance | Colorless to light amber color | Colorless to light amber color | Light amber color | Colorless | Colorless | Colorless to light amber color |
| Vapor Toxicity | Moderate irritant | Extreme concentration causes narcosis | | Irritant toxic only in large doses | Irritant cumulative toxicant Causes narcosis | Moderate irritant Extreme concentration causes narcosis |

Note 1 Not applicable
Note 2 May be the same as gasoline, or add 1.5 or 2 numbers depending on blending practice
Note 3 Values not published

SOURCES
- Bosch Automotive Handbook
- Department of Energy SAIV-1681-T1
- DOE BETC/PPS-78/3, BETC/PPS-78/5, BETC/PPS-78/4
- Ethyl Alcohol Production and Use as a Motor Fuel by J K Paul
- Marks Handbook
- Methanol Technology & Application in Motor Fuels by J K Paul
- National Technical Information Service PB 289 115
- SAE Handbook
- SAE Information Reports #J312, J313, J1297
- SAE Paper 740 599
- SAE Publication Alcohols as Motor Fuels

A considerable amount of confusion exists in the literature pertaining to the emissions resulting from methanol combustion. In general, the lower flame temperature and higher latent heat of vaporization of methanol reduces the $NO_x$ emissions in comparison to that produced by gasoline fueled engines. Since methanol is readily oxidized to aldehydes, the aldehydes are found to increase dramatically.[5,6,7]

The influence of equivalence and compression ratios, spark timing and fuel delivery mode on the performance of a methanol fueled single cylinder engine has been investigated.[8] Results of this study indicate that increased compression ratio values improve the performance of the engine, while at the same time these improvements are accompanied by increased $NO_x$ emissions in the exhaust gases. Methanol fueled engines can be operated over a much wider range of equivalence ratios than can be achieved with gasoline.[9]

Where as methanol is approximately comparable to gasoline with respect to exhaust emissions, such as CO and hydrocarbons and efficiency, methanol is worse in relation to engine starting. The cold starting problems associated with methanol use as engine fuel are directly related to its low vapor pressure and high latent heat of vaporization. (See Figure 1.1.) Reduction of the intake air temperature below 70°F results in increased engine starting difficulty. Temperatures below 5°C will not produce the necessary lean limit vapor pressure and engine starting is almost impossible without the addition of ignition improvers to neat methanol.

### 1.2.2 Neat Ethanol

The viability of ethanol as an alternate fuel depends on its availability. The primary source of ethanol is from

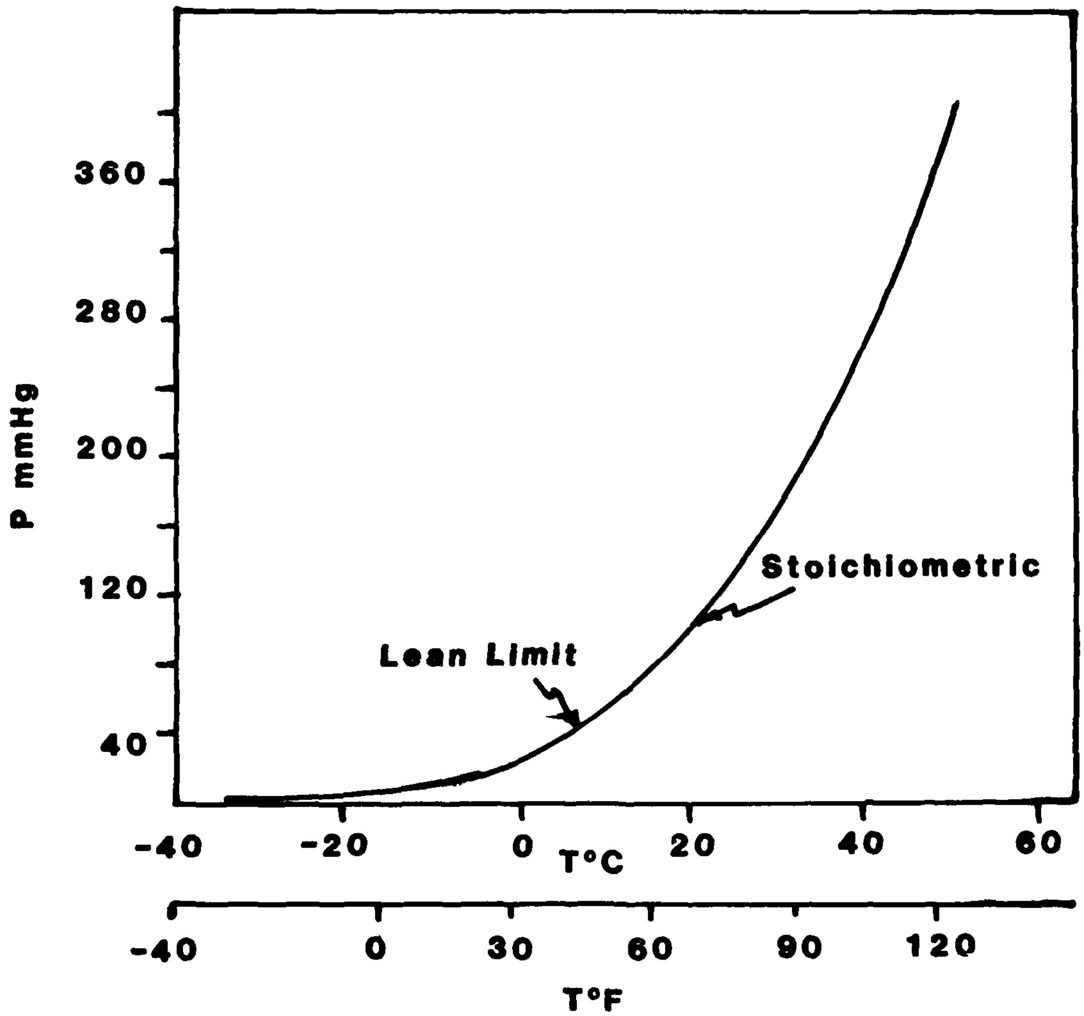



Figure 1.1 Vapor pressure versus temperature plot for methanol.

the fermentation of carbohydrates. Until recently, ethanol has not been extensively considered as a competitive fuel. However, the Brazilian efforts have greatly advanced ethanol's potential as an alternate fuel. The efficiency and exhaust emissions from a single cylinder engine fueled with ethanol has been recently presented.[10] Results are similar to those discussed for methanol. The thermal efficiency is found to increase by approximately 3 percent, $NO_x$ is observed to decrease and hydrocarbon (HC) and CO emissions are found to be comparable to those obtained for gasoline. Increased engine compression ratio, 7.5 to 18, results in a dramatic increase in all of the above characteristics. However, aldehyde concentrations in the exhaust gases are found to be much higher than for gasoline. Since the exhaust temperature is reduced with alcohol combustion, it is postulated that the latent combustion of aldehydes in the exhaust gases is dramatically reduced. This reduced combustion of aldehydes results in elevated emission concentrations.

Since ethanol's vapor pressure is reduced in relation to methanol (see Figure 1.2), cold starting is expected to be even more difficult than observed for methanol.

### 1.2.3 Alcohol-gasoline Blends

Blends of either methanol or ethanol with gasoline have not been extensively characterized. Engine performance using methanol-gasoline blends responds to equivalence ratio, spark timing and other types of engine operational variables. When comparing methanol-gasoline blends under identical conditions, it has been found that the influence of methanol on engine performance is proportional to the concentration of methanol in the mixture.[11] The effects of addition of

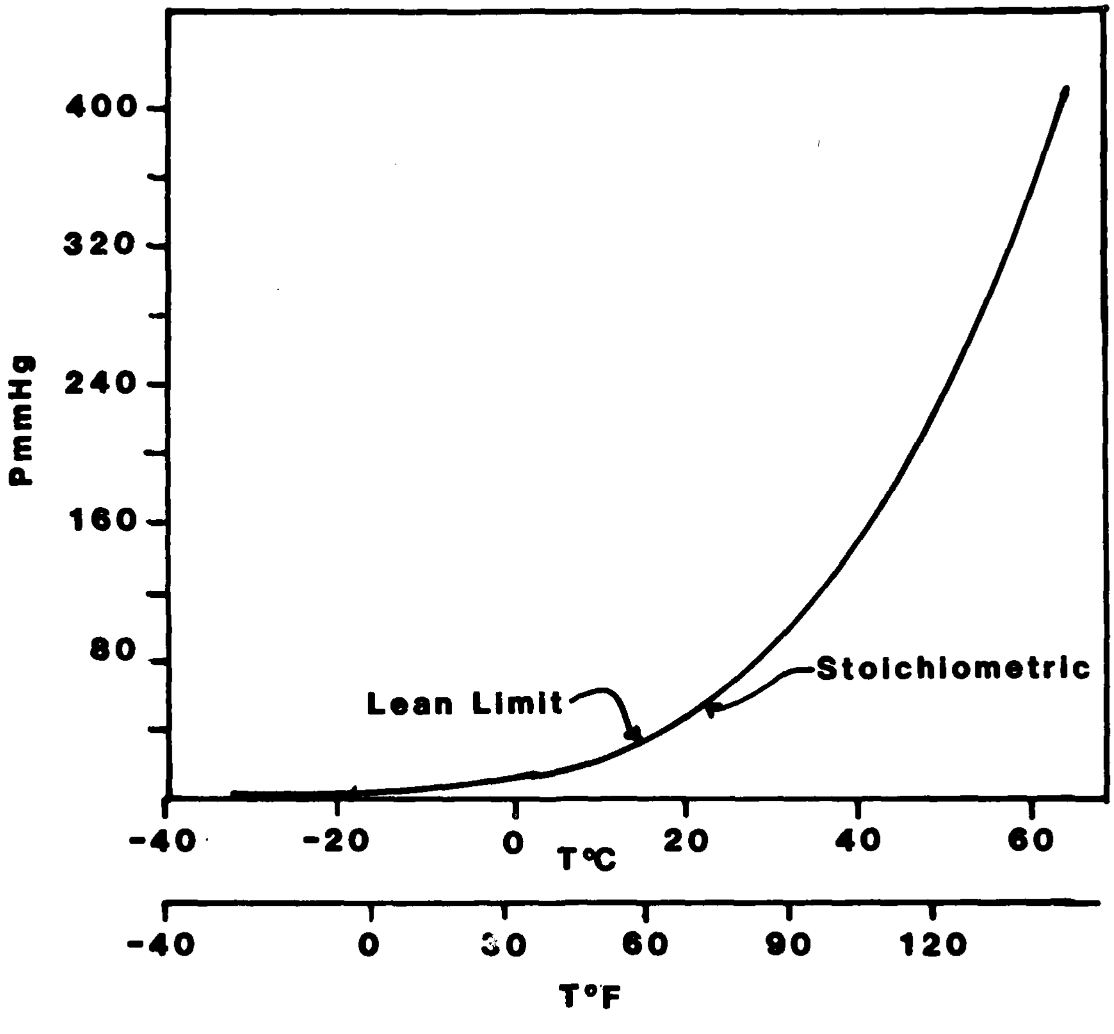



Figure 1.2 Vapor pressure versus temperature plot for ethanol.

ethanol to gasoline on engine operational characteristics follow similar trends as those reported for methanol-gasoline blends. However, the magnitude of these effects is not the same due to the difference in the physical and chemical properties of ethanol in relation to methanol.[12]

In summary, alcohols appear to be viable fuel alternatives for gasoline in times of shortage. The substantially different physical and chemical properties of alcohols in comparison to gasoline will tend to limit the performance characteristics of unmodified SI engines when powered by these fuels. The principal problems associated with the SI engine combustion of alcohol containing fuels are increased engine fuel consumption, increased difficulty of starting at reduced temperatures and elevated aldehyde exhaust emissions. Engine performance is improved when improved mixture preparation schemes, such as fuel injection, and increased compression ratio (12) are employed.

## 1.3 Engine Design Improvements

Considering the immense reserves of raw materials from which alcohol can be produced, the solution to the above problems associated with the engine combustion of alcohol containing fuels should be addressed. Major modification of the present engine designs will improve the engine performance with alcohol derived fuels. Two SI engines which utilize direct cylinder injection of fuel are currently suggested as future multi-fueled engines. The FORD PROCO concept (compression ratio = 7.5) utilizes early injection of the fuel to create a near homogeneous fuel/air mixture in moderately high cylinder air swirl. The Texaco (TCCS) engine concept (CR=12) employs late direct cylinder fuel

injection timing to create a stratified, highly swirling fuel/air mixture. At present, the Texaco concept appears to be gaining support for future engine applications. A number of non-SI engine designs have been suggested as multi-fueled concepts of the future. The most popular concept appears to be a multi-fueled diesel and the Stirling Engine designs. The Stirling Engine has been studied for a number of years; however, several material and design problems currently exist which limit its performance.

## 1.4 Engine Ignition Concepts

Major modifications to the design of the gasoline engine to produce better performance with alcohol containing fuels can only occur in new engine containing equipment. Considering the excessive time period for military certification of new concepts, and the lengthy withdrawal and retirement times for vehicles and generator sets containing these older engines, another near term solution must be found to solve the cold starting and high specific fuel consumption problems associated with alcohol use without major modification to the present engine designs.

A rudimentary examination of each of these alcohol related combustion problems indicates that an improved mode of ignition can potentially improve the overall performance of engines powered using alcohol containing fuels. The typical SI engine currently uses either a magneta or inductively charged coil to induce cylinder combustion. With the addition of the spark plug, this system represents a single point ignition system. This point ignition creates a spherical combustion flame front which propagates radially across the cylinder. Since the conventional SI engine has been designed

for gasoline, the combustion chamber shape, air flow and piston crown have not been optimized to match the single point ignition characteristic of alcohols and their blends. The problems associated with alcohol combustion in these conventional engines are due to this fact. The reduced heat release from the combustion of alcohols favor incomplete combustion of the mixture due to the radial flame growth being quenched by the heat loss to the combustion chamber walls. This effect would be significantly enhanced by lean fuel/air equivalence ratios. Due to the reduced volatility and high latent heat of vaporization, inhomogeneous fuel/air mixtures will be produced in the engine cylinder. Since ultra-lean and ultra-rich fuel/air pockets will be uniformly distributed throughout the cylinder charge, the typical radical flame propagation through these uneven distributions of fuel and air will also result in incomplete combustion of the charge.

An improved ignition source is required which enhances the ignition process and insures the total combustion of the mixture. It is important that the ignition system will produce optimum combustion characteristics independent of the fuel/air equivalence ratio. Since lean fuel/air mixtures are created when unmodified carbureted engines are fueled with alcohol containing fuels, it is important to insure the mixture not only burns, but consumes the entire charge in a time scale sufficient to guarantee maximum chemical to mechanical energy transfer.

Lewis and Von Elbe[13] have extensively studied the ignition requirements of various fuel/air mixtures. In general, these studies have shown the amount of energy required

to successfully ignite the combustible mixture is substantially increased as the equivalence ratio is decreased from the stoichiometric value. Since the conventional engine has been optimized for near stoichiometric operation, the typical spark energies are low and of the order of 10 to 30 mj. The use of higher energy delivery ignition systems has been shown to improve the lean operation of a single cylinder engine.[14] Three types of high energy ignition systems are presently considered for SI engine applications. These are multi-spark, extended duration and plasma jet ignition. These systems employ as much as 100 to 200 times the normal amount of spark energy used in conventional ignition systems.

The multi-spark and extended duration ignition systems are similar in operation and are basically dependent on the mixture turbulence to sweep fresh mixture into the spark plug region. The multi-spark ignition concept uses a series of high voltage discharges spread out over a period of up to 10 milliseconds. Each successive spark potentially ignites fresh mixture. The extended duration system, employed in both the FORD PROCO and Texaco engines, is an improved version of the multi-spark concept. In this system, a single discharge is maintained during a period of approximately 10 milliseconds. The improved operation of this system is due to the continued presence of the discharge. The extended duration spark has proven to be very desirable for ignition in high swirl engines, due the very high degree of mixture motion.

PJI is a technique which until recently has had little attention.[15] This ignition system performs in a very different mode of operation as compared to the conventional

spark plug and the high energy systems discussed above. In this ignition concept, a high voltage and high current discharge is produced in a small cavity. The small cavity is formed by the volume vacated when the electrode is withdrawn slightly into the ceramic body. The electrical discharge produced across the cavity heats the gases in the cavity and results in an instantaneous pressure buildup inside the cavity. A metal plate having a small centrally located hole is placed across the exit of the cavity. The pressure differential across this hole creates a high velocity jet of hot gases that are ejected into the combustion chamber. Since the hot jet of gases acts similar to a torch for ignition of the chamber charge, a very rapidly spreading flame occurs. Recent technology advances have improved this concept to a sufficient level of sophistication for use in general SI engine applications.

When considering the available means of improving the operation of the SI engine with alcohol containing fuel mixtures, the PJI concept is the most promising, especially for improving the lean fuel/air performance characteristics of alcohol fueled engines. PJI is attractive for SI engine use because of:

- its adaptability to most engine designs,
- its ability to increase flame spread in the combustion chamber,
- its utility for ignition of lean fuel/air mixtures.

## 1.5 Background of Plasma Jet Ignition

PJI has been extensively studied, since it was first proposed by Waterson.[16] The physical and chemical processes

responsible for PJI have been extensively characterized in atmospheric pressure combustion chamber experiments.[17,18,19] The findings of these investigations indicate that the performance of the jet, i.e., jet penetration, lean limit ignition characteristics and power requirements, is controlled by the physical dimensions of the internal cavity and the orifice diameter. Parametric studies[19] have further identified the size characteristics of the cavity in relation to the orifice diameter which produce optimal combustion at both atmospheric and engine representive pressures. The role of chemical effects in PJI have been investigated. High concentrations of radical species produced upon addition of extra fuel to the PJ cavity gases have been identified as a viable means of improving the lean limit ignition energy requirement of PJI.[20,21] These fueled cavity experiments have been extended to include solid fuels such as the inorganic metal hydrides and paraffin.[19]

Several plasma jet (PJ) plug designs have been proposed [20,22,23,24] for engine applications. Of these designs, three geometries have been tested in single cylinder engines.[22,23,24] Results of these independent studies generally conclude that PJI influences the single cylinder performance and operating characteristics in the following:

1. Exhaust temperatures are greatly reduced with PJI.
2. Output power of the engine is increased relative to the original equipment spark ignition.
3. Specific fuel consumption is reduced.
4. Lean operational limits are extended.

These engine results suggest that PJI is a very viable and versatile ignition source. Unique to the design of the

PJ plug is the ability to tailor the physical characteristics of the plug to control the extent, the position and rate of mixture ignition and combustion. The amount of energy required to power the engine compatible plug has been substantially reduced[19] such that self contained, compact ignition systems are presently feasible. These attributes are highly desirable for the ignition of alcohol containing mixtures, especially in terms of the problems associated with the resulting engine performance.

### 1.6 Scope of Present Efforts

In this study, the feasibility was examined for using PJI to ignite neat and gasoline blends of both ethanol and methanol. Combustion chamber experiments, optical engine measurements and operating engine performance data and exhaust emissions were used to characterize the influence of the mode of ignition on the rate of combustion, product gas distribution and overall ignitability of the neat alcohols and their gasoline blends. Three types of ignition mechanisms were compared, conventional magneto-spark ignition, high energy, conventional spark plug ignition and plasma jet ignition. The results of these determinations were used to characterize the following characteristics of alcohol containing fuels when ignited using PJI:

1. Lean limits of operation,
2. Influence of engine compression ratio,
3. Improvements in mixture burning velocity.

## 2.0 EXPERIMENTAL

### 2.1 Fuels

Three different fuel types were used in these measurements; solvent ethanol, neat methanol and 98 research octane unleaded gasoline. Due to alcohol restrictions, absolute (100%) ethanol could not be purchased. A solvent grade ethanol was obtained which was denatured and contained a combination of 1% of each of the following: ethyl acetate, methyl-isobutyl ketone and heptane. This solvent ethanol readily dissolved in all types of gasoline. The resulting mixtures appeared to be stable with respect to moisture and extended periods of standing. Absolute research grade methanol was used without incorporation of any additives. Mixtures of leaded gasoline with 30% methanol were very unstable and separated into a suspension after passing the fuel through the fuel injector. A similar effect was observed with the unleaded gasoline, if the mixture was allowed to stand exposed to the air (moisture) for extended periods. However, fresh mixtures of methanol and unleaded gasoline were stable and did not separate. Thus, unleaded gasoline was used throughout these measurements to insure mixture stability.

The typical preparation procedure involved mixing the appropriate amount of a particular alcohol blend before each measurement. Any remaining fuel after the test was discarded to insure that fresh mixtures were always used.

### 2.2 Ignitors and Power Supply

### 2.2.1 Convention Spark Plug

A conventional 14mm threaded, long reach spark plug manufactured by NGK was used in this study. The OEM (original equipment manufacture) ignition system was composed of a magneto driven capacitive discharge circuit and the conventional spark plug. The electrode gap was set in this configuration at 0.63mm. The high energy ignition system (CI) used the high energy PJI power supply and the conventional spark plug. The electrode gap of the CI spark plug was extended to 2mm.

### 2.2.2 Plasma Jet Plugs

Two types of plasma jet (PJ) plugs were used in these measurements. The only difference between these two plugs was the configuration of the internal cavity. The basic design of the PJ plug is shown in Figure 2.1. The PJ plug is composed of a ceramic insulator, central electrode, metal body with retainer and internal cavity. One of the two plug types tested had a fueled internal cavity insert, and the other had an internal segmented gap insert. The epoxy encapsulated insert was used to add additional fuel to the PJ cavity gases. The segmented gap did not add any additional fuel to the PJ cavity.

Figure 2.2A represents the typical structure of the metal hydride containing cavity. An epoxy encapsulated $ZrH_2$ insert, forms the walls of the small PJ cavity. A ceramic or teflon insulator is placed between this insert and the orifice plate to inhibit self-discharge of the stored capacitor energy through the hydride insert.

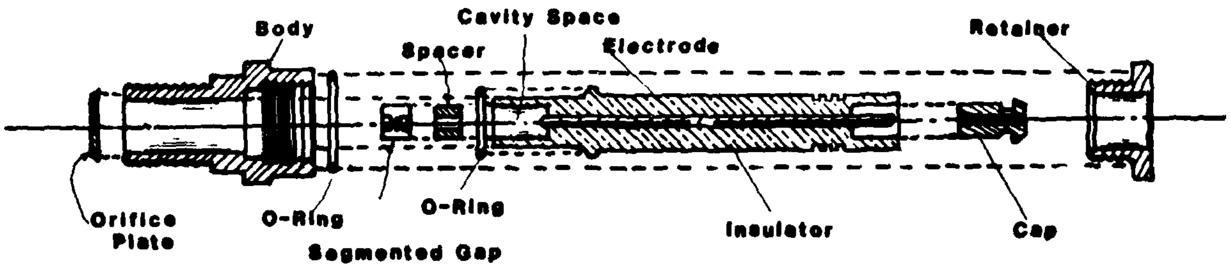



Figure 2.1 Basic design of the Plasma Jet plug.

The hydride insert is thought to add hydrogen to the PJ cavity gases by the following mechanism. The decomposition of the metal hydride to form metal plus hydrogen is temperature dependent. In the case of zirconium hydride, this hydride decomposes rapidly above 420°C. As described in Chapter 1, a series of events is responsible for PJ formation. The process is initiated by a very high voltage pulse ( 35 kV) which ionizes the gases in the PJ cavity. Once the gases are ionized, a high current, capacitive discharge occurs. This discharge rapidly heats the gases inside the cavity to temperatures in excess of 6000°K. The resulting hot gases eventually collide with the walls of the cavity. These collisions cause the wall material (hydride) to heat up. After several firings of the plug, the epoxy encapsulated insert material slowly deadsorbs hydrogen which passes into the PJ cavity. This heating process is further enhanced by the presence of combustible gas mixtures inside the PJ cavity which is representative of the experimental conditions. The net result of this chain of events is hydrogen enrichment of the PJ gases. This hydrogen enrichment has been shown to be a superior means of enhancing the PJI performance.[20]

Figure 2.2B displays the cavity structure for the segmented gap plug. This design represents an unfueled (passive) cavity geometry. The only fuel in the PJ cavity is provided by the fuel/air mixture diffusing into the cavity from the cylinder. The unique feature of this design is the segmented, internal gap. This gap is composed of a series of metal rings which are isolated from the central electrode, orifice plate and each other by ceramic insulator cups. This segmented gap drastically reduces the voltage necessary to sustain the high current discharge. Figure 2.B graphically displays

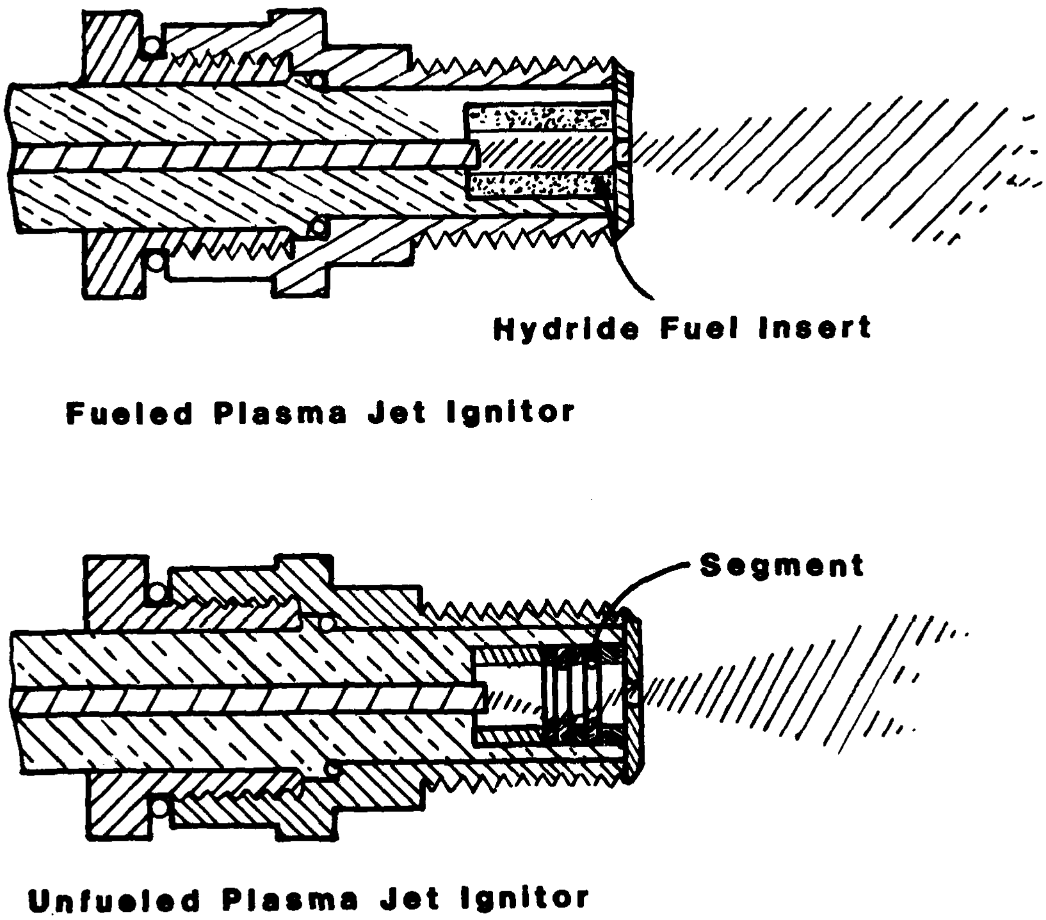



Figure 2.2 Plug-tip blowup of the Plasma Jet Ignitor.
A. Hydride fueled plug design
B. Segmented gap plug design

the discharge which occurs inside this gap and illustrates the discharge "hopping" from ring to ring. This hopping is important at elevated pressures (>10 atm.). At these pressures the previously discussed hydride cavity requires the high current discharge voltage to be in excess of 3000 V. In comparison, the segmented gap will discharge reproducibly at values as low as 700 V at these same pressures.

It was important to test both of these types of plugs, since they represent the best candidates for future PJI systems.

### 2.2.3 High Energy Power Supply

The system design shown in Figure 2.3 represents the PJ/ high energy power supply. This system operates on 3 phase 110 VAC and is capable of power output of 250 watts under continuous operation at 3000V. This power output is sufficient to operate the PJ plug or the conventional spark plug systems at 1 joule of electrical discharge energy per firing in a four cylinder engine geometry at greater than 4000 rpm. The power supply is composed of four sections, high voltage power supply, low current power supply, trigger circuit and firing circuit.

The trigger circuit is activated by an ignition start signal from the engine ignition timing unit. This incoming signal is buffered through a series of inverters. The resulting signal from the last inverter gates a one-shot multi-vibrator. The resulting pulse is then transmitted to the firing circuit where it activates an opto-SCR, $D_{13}$. This opto-SCR shorts $C_2$ to ground. A negative 280 V pulse is discharged through the primary of transformer $T_4$. A 15 kV pulse is generated at the secondary of $T_4$. This pulse

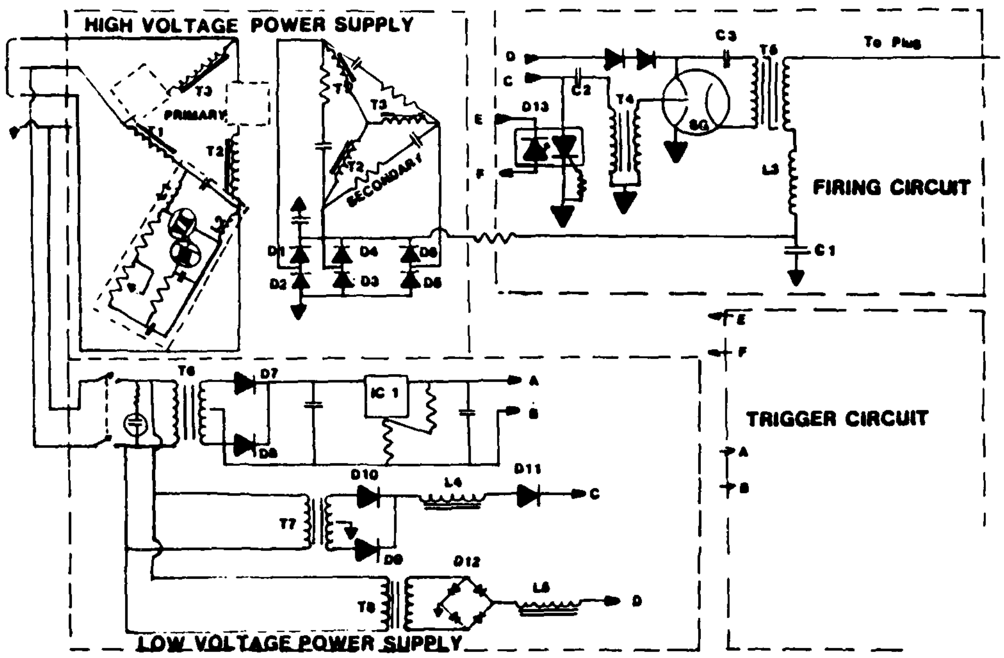



Figure 2.3 Electronic schematic of the high voltage power supply.

ionizes the spark gap, SG. Once the spark gap is ionized, $C_3$ is discharged through $T_5$. A 1000V pulse passes through $T_5$ and generates a 35 kV pulse at the secondary of $T_5$. The 35 kV pulse is transmitted to the spark plug where it ionizes the plug gap. The inductor $L_3$ chokes the 35 kV pulse from passing into the capacitor $C_1$. After the plug gap is ionized capacitor $C_1$ discharges its stored energy through $L_3$ into the plug gap. The system is recycled after the discharge in the gap is completed.

## 2.3 Combustion Chamber and Optics

Figure 2.4 displays a schematic view of the experimental apparatus employed in the combustion chamber experiments. The combustion chamber has a viewing diameter of 4 inches and a total volume of 205 cubic centimeters. The chamber is provided with four ports which are oriented at 90° intervals about the circumference. The PJ plug or conventional spark plug, piezoelectric pressure transducer, fuel injector and blank are located in these ports. The plug (PJ or spark) is located in the top port and pointed downward.

The typical operating procedure involved evacuating the chamber before each measurement. The fuel was injected into the evacuated chamber and allowed to flash evaporate. The air was introduced and the resulting mixture was permitted to equilibrate to atmospheric pressure.

Photographic images of the combustion event were recorded using a Z-plane, argon spark shadowgraph. The time history of the combustion event was obtained by sequentially varying the time delay between the firing of the PJ and the argon spark.

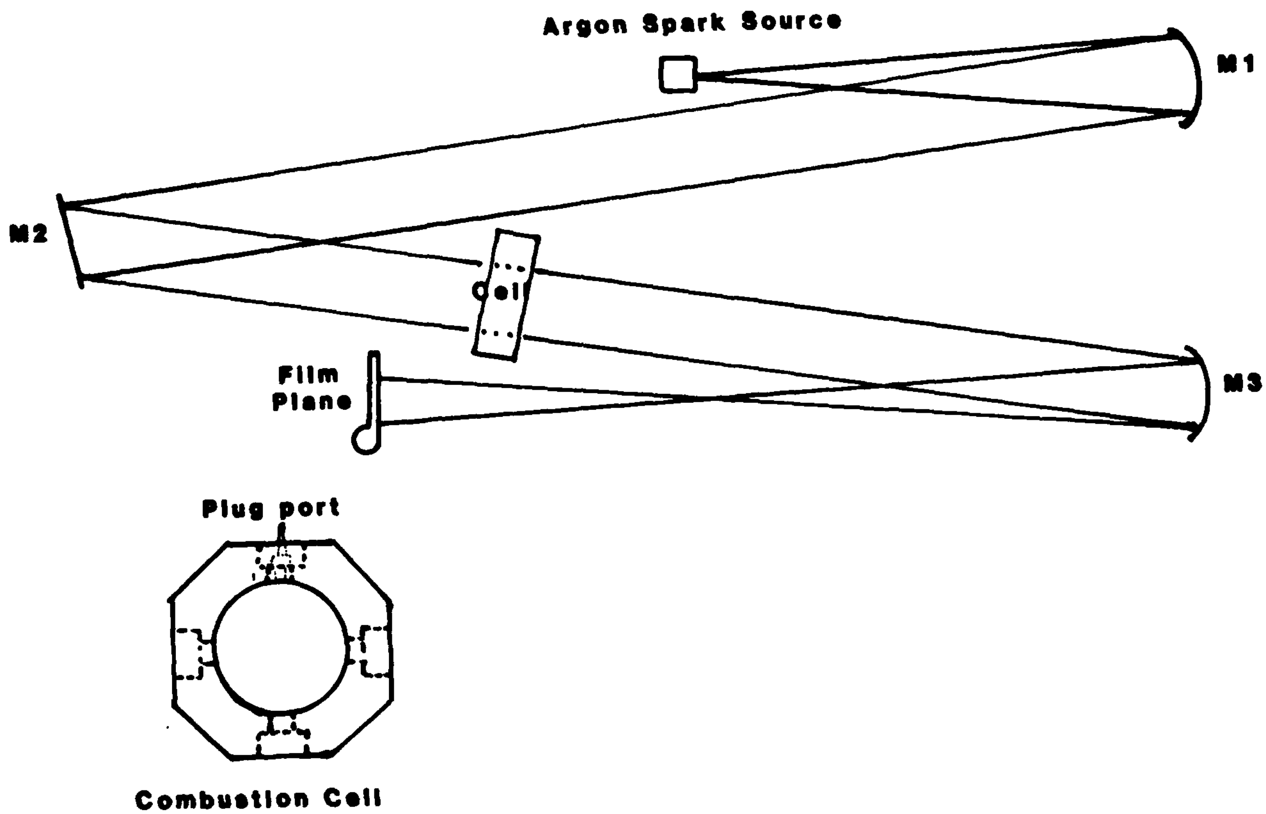



Figure 2.4 Schematic view of the combustion chamber experimental aparatus.

## 2.4 Optical Engine

Figure 2.5 displays the apparatus associated with the optical engine. The principal component of the optical engine is a Rotax model 292 two cycle engine block. The engine was motored using an inline coupled 1.5 horsepower DC electric motor. Using this setup, the engine speed can be uniformly varied from 300 to 2000 rpm. The optically accessible head is shown in Figure 2.6. A 3 inch by 1 inch thick quartz window forms the top section of the engine combustion chamber and permits a 2.25 inch viewing diameter inside the 3 inch bore of the engine. Side ports are machined into the head to allow for positioning of the spark plug and a pressure transducer. Due to the desire to maintain a reasonable compression ratio, the plug port does not penetrate into the combustion chamber. (Typically, the plug electrodes are exposed to the inside of the combustion chamber.) The optical engine head combustion chamber size is approximately twice the volume of the original Rotax head. This fact reduces the original compression ratio from approximately 8 to 1 down to 4.3 to 1. A specially ground quartz window can be inserted into the window port, shown in Figure 2.6, and the resulting geometry produces a compression ratio of 6 to 1.

The air flow into the engine was monitored using the pressure differential across a flow nozzle. The fuel was injected into a cylindrical intake channel and the throttle was always in the wide open position. The air/fuel mixture was passed through a motionless mixer before entering into the engine. This mixer helped to increase the homogeneity of the fuel distribution in the mixture. Both the intake

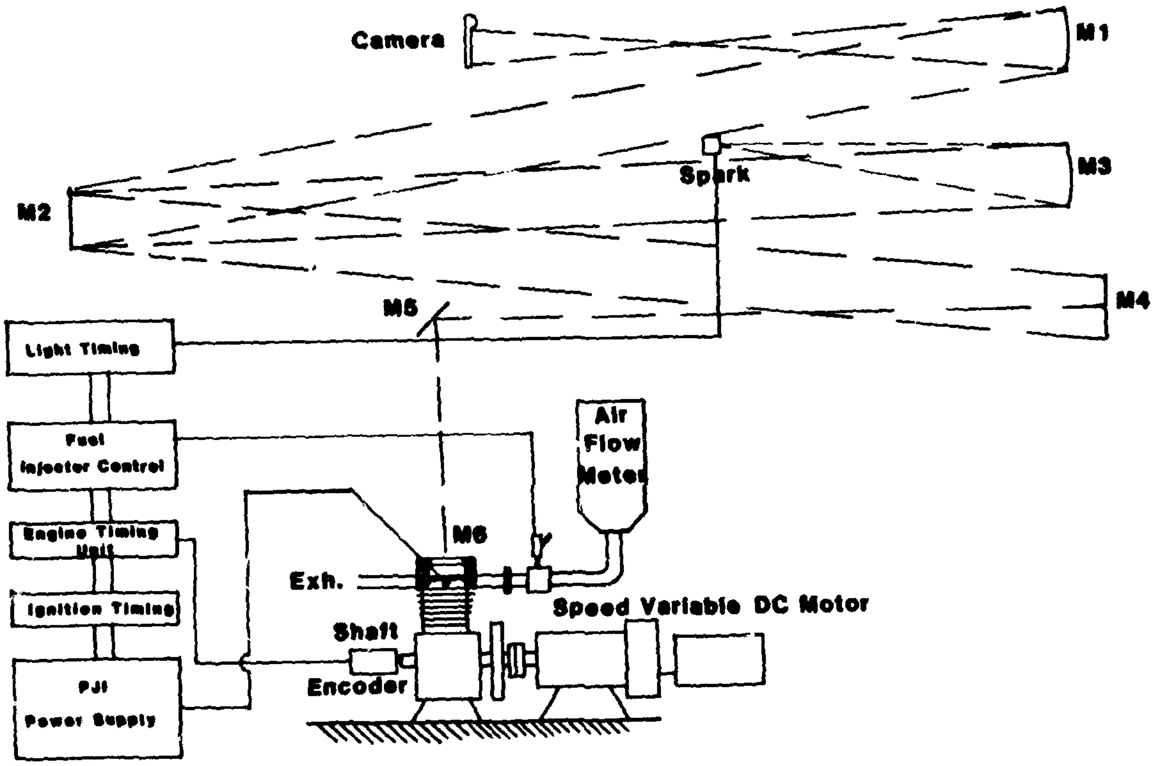



Figure 2.5 Schematic view of the optical engine and associated apparatus.

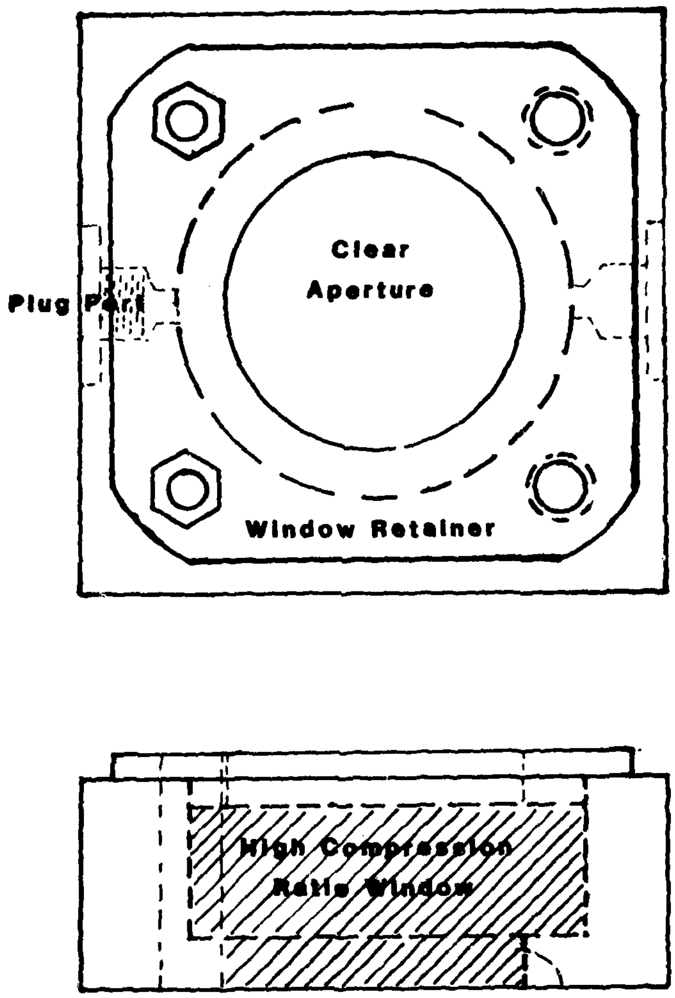



Figure 2.6 Design of the optical engine head, and high compression window.

manifold and the mixer were temperature controlled.

Figure 2.5 illustrates the electronic control system for the optical engine. A 360° resolution shaft encoder monitors the position of the crank shaft. This unit transmits a pulse for every crank degree and a top dead center (TDC) pulse. Three timing circuits monitor these pulses; the fuel injection timing and duration, ignition timing and photo timing boards. The fuel injection timing and duration circuit controls the point in the engine cycle when the fuel is injected into the air flow. The unit also controls the amount of time that the injector remains open. The injection time can be varied through the entire 360° engine cycle. Typically, the inlet to the Rotax scavenging chamber is opened at about 290° after TDC. The fuel injection is timed to coincide with this opening. The injection duration is variable over a range from 0 to 5 msec which allows for a wide range of fuel/air equivalence ratios. The ignition timing is controlled by a divider/counter circuit. The ignition timing is varied by a series of thumb wheel switches over a range of -99 to +70 degrees of TDC. A similar circuit controls the firing of the argon spark necessary to monitor the combustion event.

Figure 2.5 displays the optical setup associated with the engine. This system is similar to that used in the combustion chamber measurements. The optical paths are folded onto one another. This folding was necessary to remove the light generated inside the engine from interfering with recording of the shadowgraph. The temporal development of the combustion event was captured by the shadowgraph by varying the crank angle position at which the argon spark was discharged.

## 2.5 Single Cylinder Engine Measurements

### 2.5.1 Engine

The engine employed for these measurements is a Honda Motor Company model G400 gasoline engine. This engine is an air cooled, 4-stroke, side valved, single cylinder unit. At 3600 rpm, the engine develops a maximum of 10 horsepower and a torque of 16 ft.-lbs. The compression ratio of this engine is 6.5 to 1. The crankcase lubrication system has been modified for high volume oil flow and cooling of the oil. The stock engine is supplied with a horizontal carburetor which has been replaced by a Kawasaki fuel injection manifold. The engine intake and exhaust are coupled to surge tanks in order to damp the gas pulsations. The engine is mounted on a vibration isolated test stand. The engine drive shaft is coupled through a high resolution torque meter to a 115 VAC, 30 amp single phase generator. A load bank is used to load the engine. This bank is able to apply up to 3000 watts of load.

### 2.5.2 Engine Control and Instrumentation

An electronic control system similar to that employed with the optical engine was used to regulate the fuel injection and ignition timing. Since this engine was a 4-stroke configuration, additional circuitry was necessary to properly time the necessary control events.

The engine diagnostic instrumentation is shown in Figure 2.7. The data collection is controlled by a DEC LSI-11/23 mini-computer with disk storage capacity. The analog data is collected by a 99 channel micro-computer system (Himmelstein 6), digitized and transmitted to the mini-computer.

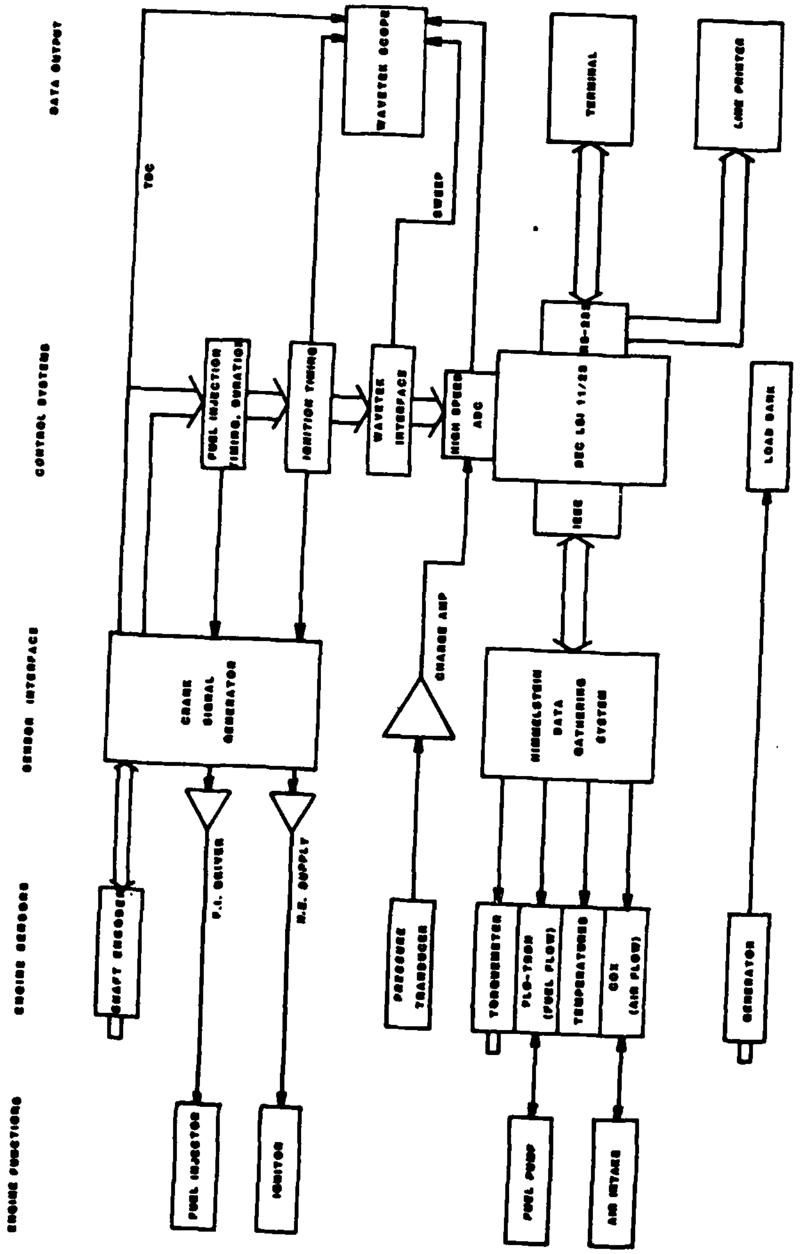



Figure 2.7 Honda engine electronic control system.

The following engine data are collected:

1. Air flow into the engine is determined by the pressure differential across a calibrated flow nozzle.

2. Fuel consumption is monitored by a temperature and pressure compensated flow meter.

3. Combustion chamber, ambient, fuel, oil and exhaust temperatures are determined using thermocouples.

4. Engine torque output and load are also monitored.

2.5.3 Test Procedure

The testing performed in this study followed a major portion of the specifications called out in the SAE (Society for Automotive Engineering) test procedures 607a and 816b. Although these procedures define the method for collection of an entire engine map, only a single rpm value was mapped. The ignition timing was set to the factory specified value of 22 degrees before TDC. The load on the engine was held constant at 1000 watts. The engine rpm was maintained at 1800 rpm. The engine performance (torque and brake specific fuel consumption) were determined for three types of ignition (OEM spark, HEI spark and PJI) and three types of fuel (gasoline, 30% methanol/70% gasoline and 30% ethanol/70% gasoline).

Hydrocarbon, aldehyde and carbon monoxide concentrations were determined in the exhaust gases using a Hewlett-Packard 5763 gas chromatograph. This system was equipped with both flame ionization and thermo-conductivity detectors. Detector response was calibrated using known concentration standards.

The typical operating procedure involved setting the fuel injection duration at a specified value. The throttle was adjusted to produce the desired rpm level. The engine was then allowed to equilibrate for approximately 10 minutes. A series of approximately 200 data sets were determined and averaged. An exhaust gas sample was collected and analyzed. The next test point was set and the procedure repeated.

## 3.0 RESULTS AND DISCUSSION

### 3.1 Burning Velocity

The engine combustion process, as previously discussed, is tailored to the combustion properties of gasoline. Alcohols are known to have different combustion properties than those of gasoline. It is also important to identify how the combustion characteristics of gasoline are modified on addition of alcohol. In terms of these combustion properties, the combustion rate is extremely important. The proper burning velocity (combustion rate) is critical to efficient engine operation because this velocity is related to the rate of chemical energy release by the charge. This rate of chemical energy release is directly proportioned to the overall chemical to mechanical energy transfer in the engine.

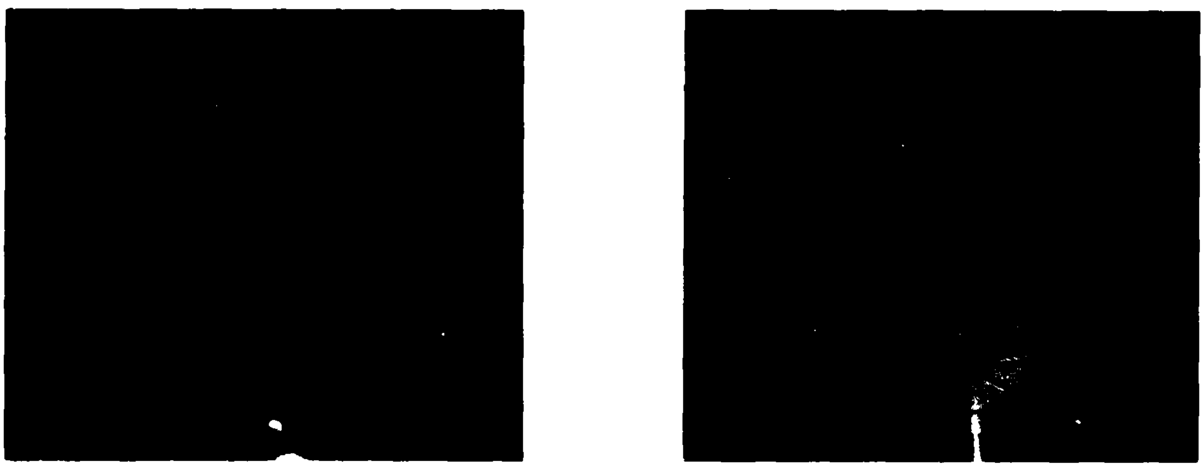
The measurements reported in this section examine how the addition of alcohol to gasoline influences the burning velocity of the resulting fuel/air mixture. Specific factors to be examined are:

1. The effect of the mode of ignition.
2. The effect of the concentration of alcohol in the fuel.
3. The influence of the initial mixture pressure.
4. The influence of mixture turbulence.

#### 3.1.1 Quiescent Mixtures

Based on the results of previous studies involving PJI, enhanced combustion rates were expected to be produced by PJI in any quiescent fuel/air mixture. However, it was important to demonstrate that PJI also improved the turbu-

Time = 5 msec



Time = 10 msec

PJI CI

Figure 3-1. Typical shadowgraphs used in the combustion chamber burning velocity analysis. Comparison of PJI and CI of 10% ethanol-gasoline blend, $\phi = 0.6$.

lence free burning velocity of alcohol containing fuel/air charges.

Quiesent mixtures are typically produced and studied in a combustion chamber. This combustion cell measurement not only simplifies the fluid dynamics but allows for baseline characterization of the basic fuels. This baseline is especially important in the case of solvent ethanol, since this solvent contains additives.

Figure 3.1 displays several shadowgraph photos typical of those used to characterize the mixture burning rates. The darkened area present in these photos is created by the hot gases resulting from combustion of the mixture. The two dimensional photographic images of the area is integrated to generate a value for the flame area. Since each photo represents a different time delay after ignition, each area value can be plotted with respect to the time delay. The slope of the line to best fit through these points can be related to the mixture burning velocity. Figure 3.2 presents the plots obtained for the alcohol containing fuels.

The data in Figure 3.2 was determined for mixtures initially at atmospheric pressure and room temperature. The mixture strength of the fuel in these measurements was 0.6 of the stoichiometric ratio for each fuel. As suggested by our previous PJI experiments, PJI was found to enhance the quiescent combustion of each alcohol and the various (10%, 20% and 30%) alcohol-gasoline mixtures. This comparison was made between the high energy ignition using a conventional spark plug (CI) and the PJ plug. Both were fired at the same energy. Since the slopes of the PJI plots are larger than the slopes obtained using CI, their mode of ignition must be

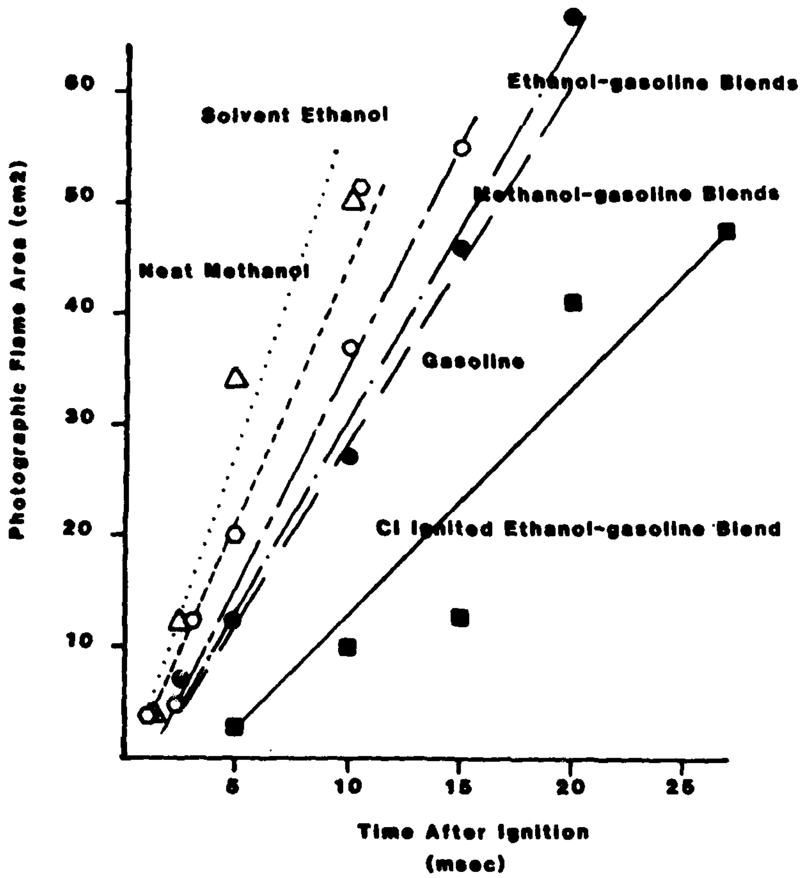



Figure 3.2 Plot of flame area present in the combustion chamber as a function of time after ignition for ethanol, methanol, (10%, 20% and 30%) alcohol-gasoline blends and neat gasoline. Equivalence ratio equals 0.6.

producing high burning velocities. The enhanced flame coverages and burning velocities again confirm the utility of PJI.

It was also apparent from this analysis that the basic alcohol/air mixtures burned more readily than either the alcohol/gasoline blends or gasoline. This result had been predicted by the literature. However, the literature suggested that methanol would have a higher burning velocity in comparison to the burning rate of ethanol. This was not observed to be true for our fuels. Ethanol and the ethanol containing mixtures were found to have larger burning rates than the methanol containing fuel air mixtures. This result had been anticipated, since the solvent ethanol was known to contain material which could potentially enhance the burning velocity of the resulting ethanol mixture.

The burning velocities of the various compositional blends of a particular base alcohol were found to have similar burning rates. When compared to previously collected data for gasoline (dashed line), all of the alcohol blends were found to have accelerated combustion rates. These results indicated that alcohol addition may act as a gasoline combustion improver. However, beyond the initial 10% addition, very little burning velocity improvement is obtained by any higher percentage of alcohol.

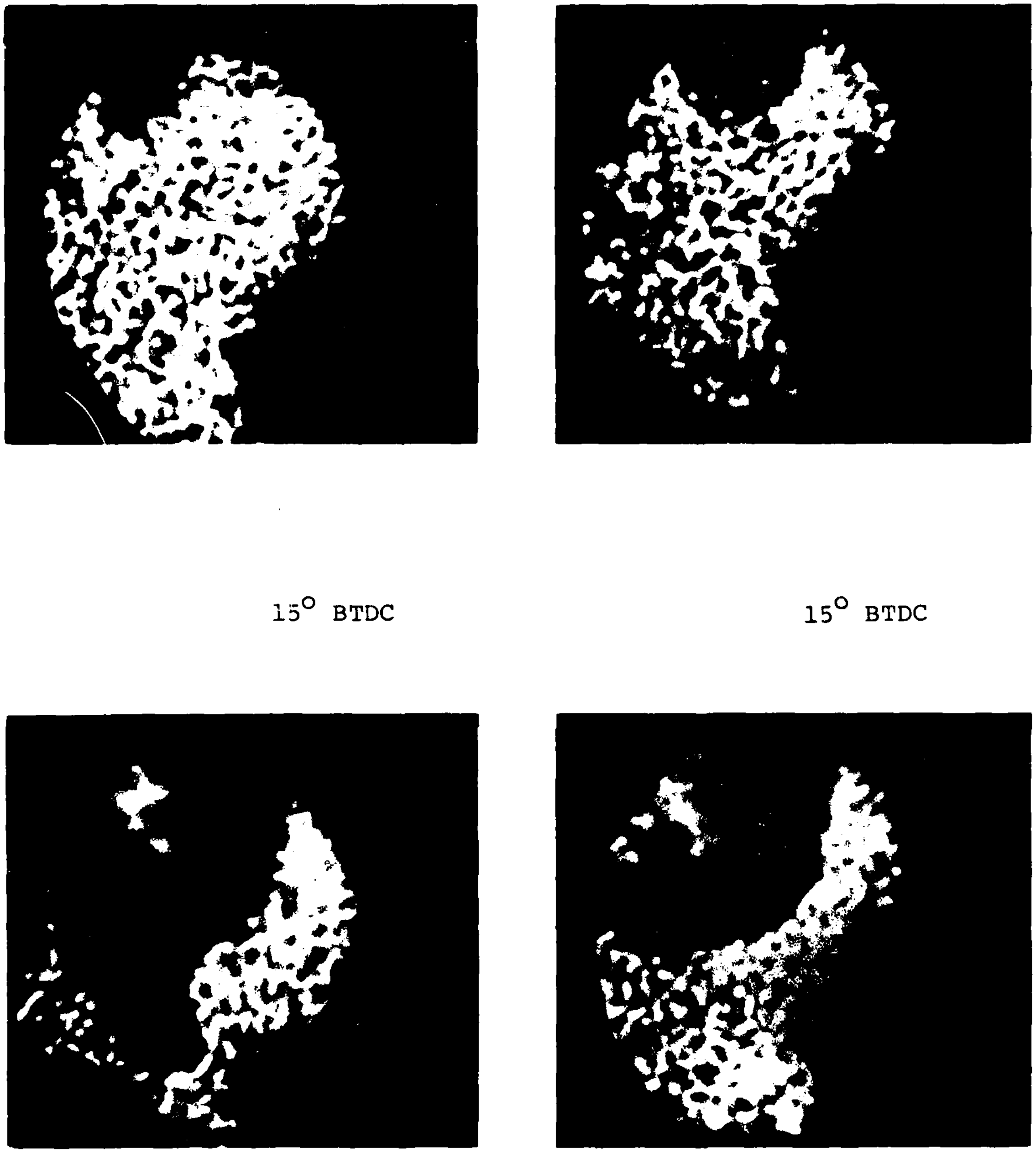
### 3.1.2 Turbulent Burning Velocities

The optional engine provides a more representative set of conditions to investigate the rate of combustion. These conditions are representative of those found in typical engines. The engine environment is characterized as a dynamic set of conditions. On intake, the gas flows into the engine through the intake port. Depending on the intake

geometry, the incoming gases will continue to move in the cylinder. The axial motion of the gas will be enhanced as the piston compresses the mixture. The flow of the gases in the cylinder will result in localized, axial mixing of the cylinder gases during the combustion process. Although the flow is seemingly localized, it will come in contact with adjacent flow regimes. The result of this contact produces enhanced combustion rates in comparison to that which occurs in quiescent mixtures. In addition to producing mixture flow, the piston also compresses the gases in the cylinder. This compression not only produces high gas pressure but results in heating of the gases. Thus, in comparison to the previous quiescent measurements, the engine combustion process is complex and represents the contribution of many factors. Figure 3.3 displays typical shadowgraphs of the combustion inside the engine.

Figure 3.4 compares the extent of combustion present in the engine as a function of crank angle for the conventional spark ignitor (CI) and the hydride fueled PJ modes of ignition. As in the quiescent experiments, the amount of stored electrical energy delivered to the plug was maintained constant and equal to 1 joule. The compression ratio of the optical engine is 4 to 1 for these measurements and results in a maximum motored cylinder pressure of 60 psi. The engine is motored at 900 rpm which results in each crank degree change being equal to approximately 180 microseconds of real time. The ignition is set to occur at 25° before TDC.

It is apparent from viewing Figure 3.4 that PJI again produces enhanced combustion in comparison to the conventional spark plug. The combustion of the cylinder mixture appears



15° BTDC | 15° BTDC

0° BTDC | 0° BTDC

30% Ethanol-Gasoline | 30% Methanol-Gasoline

Figure 3-3. Comparison of the extent of combustion present in optical engine for PJI of 30% gasoline blends (BTDC = Before Top Dead Center).

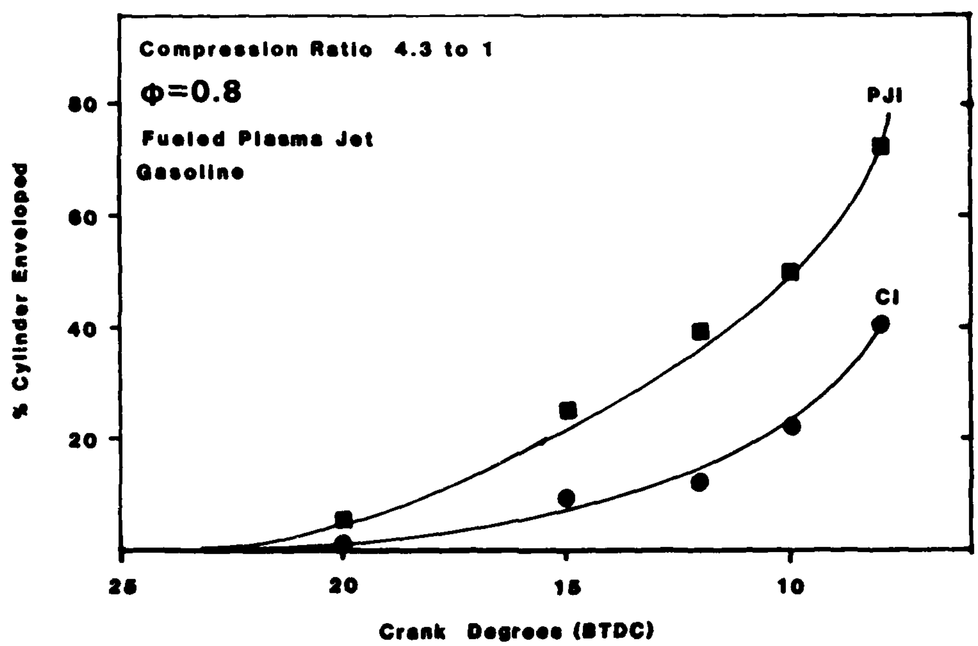
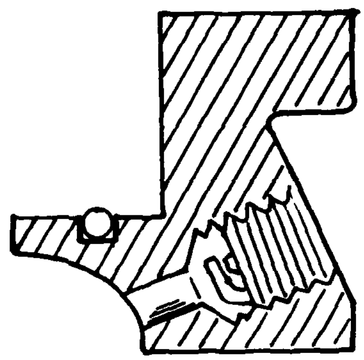



Figure 3.4 Plot of the percent of the cylinder enveloped versus crank angle for PJI and CI ignition of gasoline.

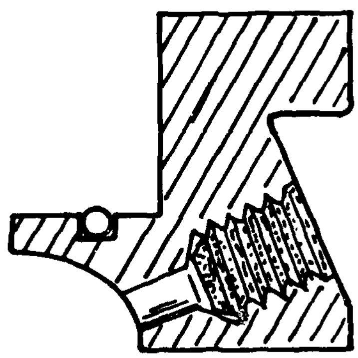
to be enhanced by at least a factor of two times the amount produced by the CI plug. This value is comparable in magnitude to the amount of PJI enhancement observed for quiescent mixtures.

The degree of enhancement found in these optical engine measurements is surprising, since the optical engine head geometry was not ideally suited for the plasma jet. This head geometry had been designed with recessed plug ports. Figure 3.5 illustrates a blowup of the plug ports in the optical engine. Figure 3.5A displays the structure of the port when the conventional spark plug is in place. Once the combustion is ignited, the flame builds inside the recessed chamber and then propagates down the interconnecting channel to the combustion chamber. The pressure buildup inside the recessed chamber will produce some jetting of combustion gases out of the port as the flame propagates along the interconnecting channel. This configuration is similar to the plasma torch. Figure 3.5B depicts the plasma jet plug in this port. Due to the difference in the plug design, the plasma jet is ejected directly into the interconnecting channel. Combustion chamber measurements have shown that the jet penetration is approximately 3 cm for this plug geometry and energy input. Considering the extent of chamber turbulence and the distance over which the jet must travel, one would not expect deep penetration of the jet into the chamber.

PJI enhancement has been identified as being due to two factors of the fluid dynamic effects, such as induced turbulence and mixing by the jet, and chemical effects, the injection of reactive species into the mixture. Considering the optical



**A) Conventional Spark Ignitor**



**B) Plasma Jet Ignitor**

Figure 3.5 Plug port design
A. With CI Ignitor in place, and
B. With the PJI Ignitor

engine geometry, the fluid dynamic effects can not be producing the observed enhanced burning velocity. Thus, the chemical effects are shown to be responsible for producing the observed burning enhancement. This is further substantiated by the fact that the hydride fueled PJ was used in these measurements. The hydride fueled PJ has been shown to chemically enhance the burning velocity of propane mixtures in relation to the unfueled PJ.

In order to confirm this chemical enhancement, a series of measurements were repeated using the segmented gap PJ plug. The segmented gap plug does not have an additional cavity fuel source. The ignition produced by the segmented gap PJ did prove to enhance the burning rate of the test mixture in relation to the CI plug. However, the magnitude of this enhancement was approximately one-third the value obtained for the fueled PJ. These findings confirm the advantage of using a fueled PJ.

Figures 3.6 and 3.7 compare the CI and fueled PJI modes of ignition for the 30% alcohol-gasoline blends. The PJ is found to also produce enhanced burning of the alcohol mixture. A factor of three increase in the burning rate is found for PJI. Figure 3.8 compares the PJ ignited results for the alcohol blends and neat gasoline. This comparison shows that the burning velocities of the alcohol blends are substantially less than gasoline. The fact that combustion is present at 20° before TDC in the case of gasoline indicates that there is a considerable ignition delay associated with the alcohol containing blends. This is contrary to the results observed in the quiescent mixtures where the alcohols are observed to improve the burning rate of gasoline.

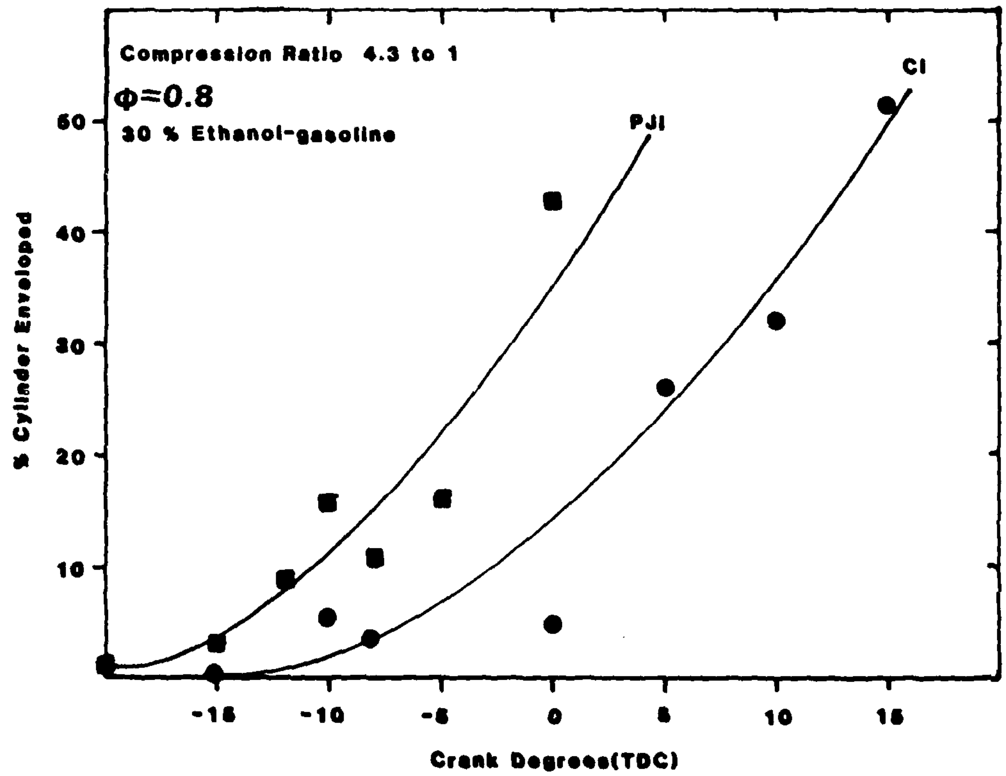



Figure 3.6 Extent of combustion present in the cylinder as a function of crank angle for PJI and CI ignition of 30% ethanol-gasoline blend.

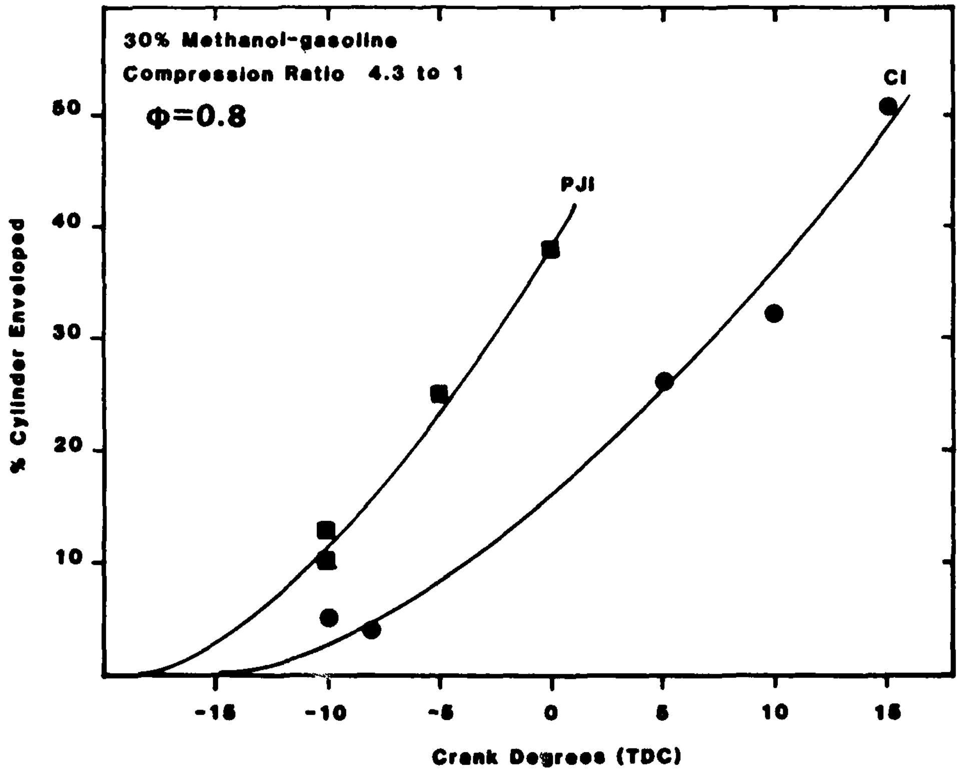



Figure 3.7 Extent of combustion present in the cylinder as a function of crank angle for PJI and CI ignition of 30% methanol-gasoline blend.

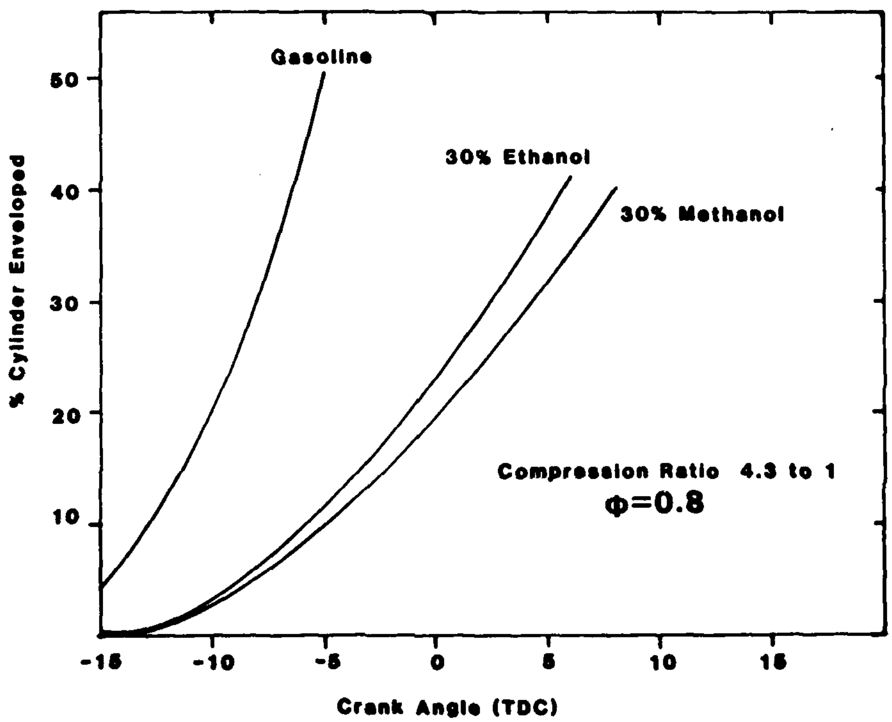



Figure 3.8 Comparison of the extent of cylinder ignited as a function of crank angle for gasoline, 30% methanol-gasoline and 30% ethanol-gasoline.

There are potentially two factors which are responsible for the observed trends.

1. The temperature dependence of the combustion rate of gasoline is greater than that of the alcohol containing blends, or

2. The pressure exponent of the burning velocity of gasoline is greater than that of the alcohol blends.

The global rate constant (k) of any set of chemical reactions can be expressed in terms of an Arrhenius equation:

$$k = A\,e^{-\Delta E/kT} \qquad (1)$$

where A is the Arrhenius constant, $\Delta E$ is the activation energy, $k$ is the Boltzman constant and T is the temperature in °K. Employing this analysis, the $\Delta E$ for gasoline combustion is less than that for the alcohol blends. This higher value of the activation energy may be substantiated by the fact that the octane number of gasoline is increased by addition of alcohol to gasoline. Since the octane number evaluates the fuel's ability to resist pre-ignition, the alcohol addition must be influencing the energy required to spontaneously ignite the mixture. The elevated energy may also be responsible for the decreased burning velocity.

Unfortunately, the above fails to explain the observed low pressure results. The mixture pressure can also influence the burning velocity of the mixture. Lewis[25] has shown that the burning velocity of a fuel/air mixture is a power dependent function of the pressure and is given by:

$$\frac{Su_a}{Su_b} = \left(\frac{Pa}{Pb}\right)^n \qquad (2)$$

where $Su_a$ and $Su_b$ are the burning velocities at $P_a$ and $P_b$ and n is the power dependence. A negative value of n results in the burning velocity decreasing with pressure. (Propane at $\emptyset = 0.7$ has a value of -0.2 for n.) Previous high pressure experiments with gasoline and propane indicate that the burning rate of gasoline does not significantly decrease at elevated pressures. Thus, the addition of alcohol to gasoline may act to reduce the value of n.

The present experiments can not resolve the observed results. However, these results suggest that the ignition timing should be retarded with respect to the value used with gasoline when alcohols are added to the gasoline. The retarding of the ignition will compensate for the decreased burning velocity and ignition delay.

Figure 3.3 represents the observed flame in the optical engine. The shape of this flame is similar to the shape determined for point ignition in quiescent mixtures. The flame appears to propagate radially away from the site of ignition. However, the leading edge of the flame is not as symmetrical as the quiescent flame shape. This wrinkled appearence is caused by the axially directed turbulence. Since there are a number of factors influencing the burning rate, the optical engine burning velocities are expected to exceed those in quiescent mixtures, and indeed, the optical engine measurements confirmed this increase.

In summary, PJI proved to be a superior ignition source for both quiescent and turbulent fuel/air mixtures. Both the segmented gap and the hydride fueled plugs yielded enhanced combustion in relations to the CI mode of ignition. This enhanced combustion was found for all fuel/air mixtures investigated.

The one-atmosphere results indicated that small additions of alcohol acted to enhance the combustion rate of gasoline. However, the opposite trend was observed in the optical engine results. The optical engine results further indicated that the addition of alcohol to gasoline increased the mixture ignition delay. These findings suggested that the ignition timing should be retarded with respect to that of gasoline to compensate for this ignition delay.

### 3.2 Lean Limit Studies

PJI has been previously shown to extend the lean ignition limit of many gaseous fuels, such as methane, propane and butane. It was of interest to demonstrate that PJI would also extend the lean limit of alcohol containing fuels. Since the lean ignition limit of the fuel/air mixture has been related to the engine's starting characteristic, these studies were important to define how the type of ignition system influenced the "cold" starting of the engine when fueled by alcohol containing fuels.

The lean limit is another property used to describe the combustion of a fuel/air mixture. The lean combustion limit is characterized in a combustion chamber as the smallest amount of fuel which will produce flame propagation in the fuel/air charge. In a motored engine, the lean limit is the lowest equivalence ratio below which ignition of the fuel/air mixture will produce no noticable change in the observed motored pressure trace. As the lean limit is approached, the ignited combustion of the mixture usually persists into the exhaust manifold. This situation is usually associated with further combustion of the residual exhaust gases already present in the exhaust system. This combustion

of the exhaust gases commonly produces a loud explosion. As this limit is approached in the operating engine, very rough engine operation with erratic cylinder pressurization, elevated exhaust hydrocarbons and low torque output are produced.

The lean limit can be related to the engine starting conditions. In order for the engine to motor itself, the chemical energy released must produce sufficient pressure in the cylinder to force the piston back down the cylinder. In the warm engine, the lean limit amount of fuel delivered by the fuel delivery system will correspond directly to vapor composition due to complete vaporization of the fuel. In the cold engine, this limiting amount will not be equal to the amount of fuel initially introduced into the manifold, since the low wall temperature will cause incomplete vaporization of the fuel. The cold engine usually requires increased throttling of the engine to provide additional enrichment of the fuel in the charge.

One of the primary objectives of this investigation was to determine how the cold starting of the engine was affected by the use of various ignition systems. In this section, the influence of the mode of ignition on the lean limit of the optical engine will be examined. Since the optical engine is a motored engine, the lean limit is defined as the lowest amount of fuel which, when ignited, will produce a detectable pressure excursion above the pressure obtained by simply motoring the engine.

Five fuels were examined in these measurements, neat gasoline, neat methanol, solvent ethanol and the 30% blends of alcohol and gasoline. The lean limit was determined for

two types of ignitors PJI and CI, which were fired using 1 joule of input electrical energy. The segmented gap PJI was used throughout these measurements, since the hydride fueled plugs would not withstand repetitive firing. The intake manifold temperature was externally heated above 70° F for the warm ignition measurements and chilled for the cold limit measurements.

#### 3.2.1 Warm Limit

Figures 3.9 and 3.10 show the results obtained by these measurements. The limit with PJI was constantly below the limit obtained with the CI source. This was true for all fuels and ignition timing settings. In addition to the extended lean limit, PJI produced a uniform loss of pressure in the engine as the mixture was leaned. In comparison, the CI mode of ignition produced a delayed pressure pulse which resulted in exhaust backfire. (See Figure 3.11.)

In terms of the influence of alcohol on the lean limit, the addition of alcohol to gasoline extended the range of equivalence ratios over which the engine would operate. In fact, gasoline has the highest lean limit, whereas ethanol and methanol had the lowest values.

The extremely low lean limit for methanol is suspicious, since this value is below the reported lean limit. This low value is possibly due to the fact that the optical engine is a two cycle engine. A two cycle engine does not ingest the manifold fuel/air mixture directly into the cylinder. The intake gases initially are drawn into the scavenging chamber of the engine. During the exhaust stroke the fuel/air mixture in the scavenging chamber is pushed into the engine cylinder. If the temperature of the walls of the scavenging

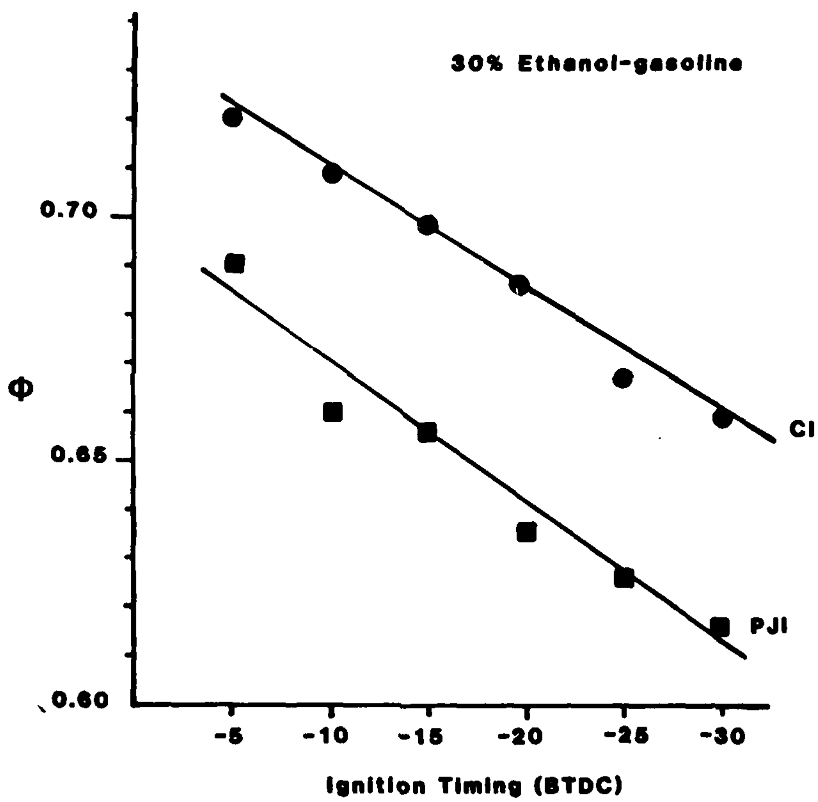



Figure 3.9 Lean misfire limit as a function of ignition timing for 30% gasoline blends of ethanol and methanol.

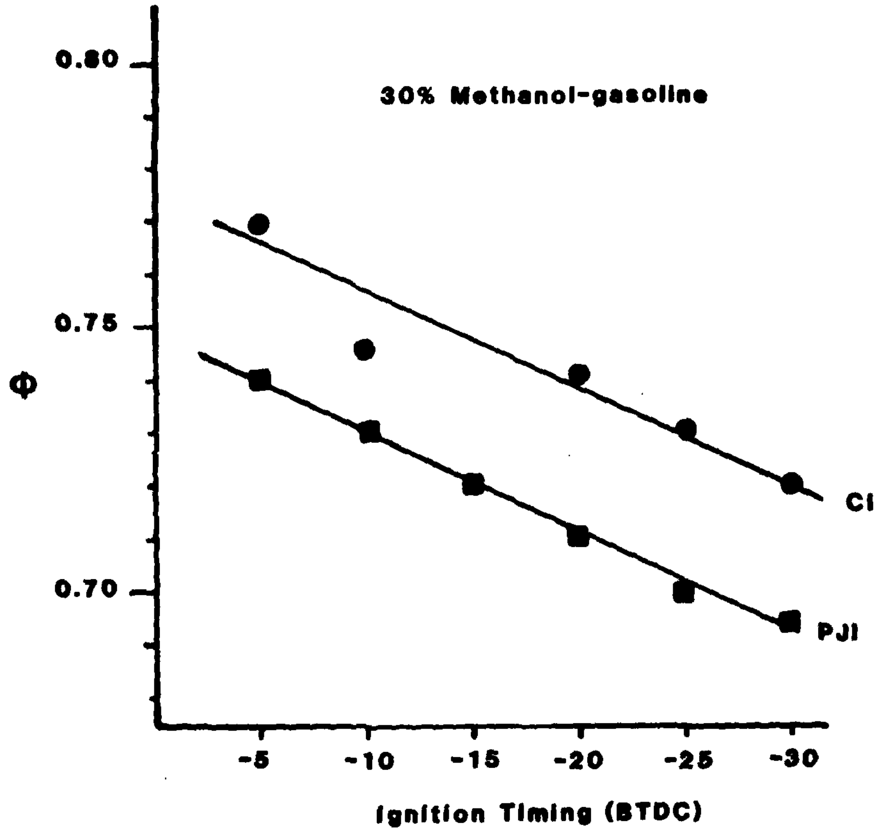



Figure 3.9 Lean misfire limit as a function of ignition timing for 30% gasoline blends of ethanol and methanol.

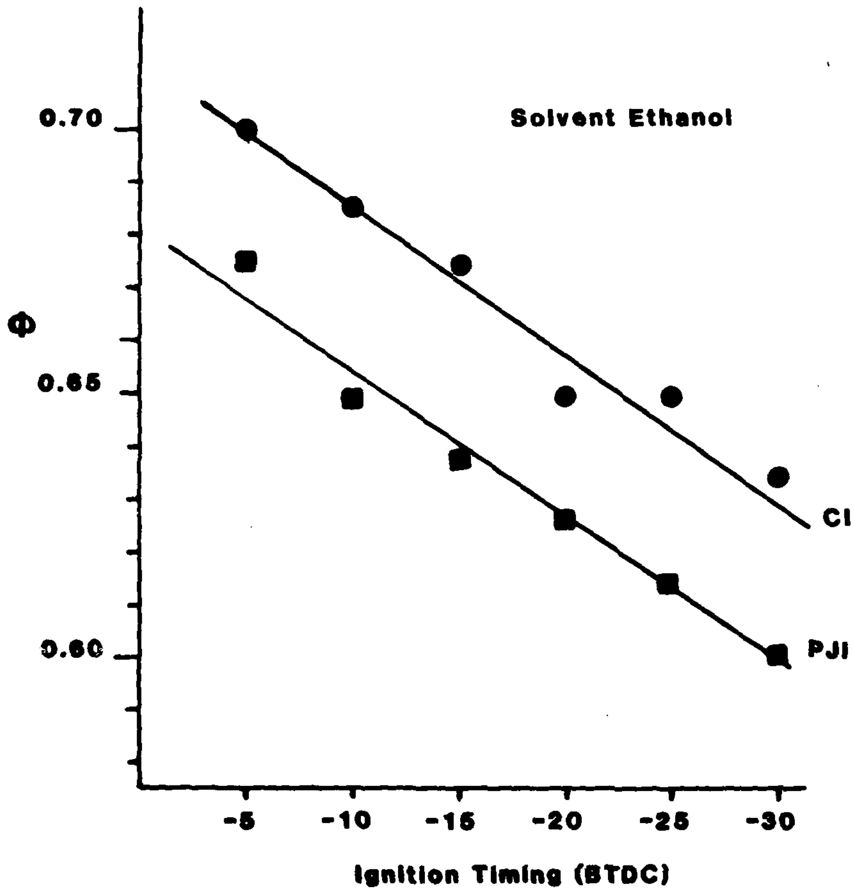



Figure 3.10 Lean misfire limit as a function of ignition timing for neat methanol and solvent ethanol.

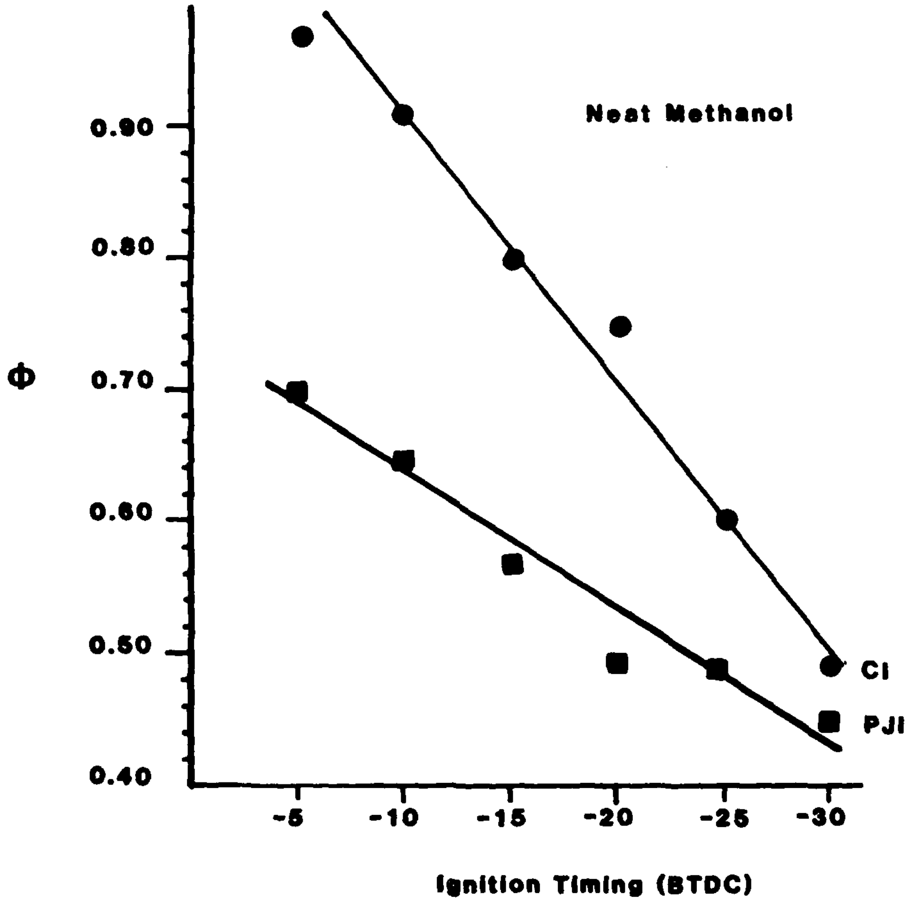
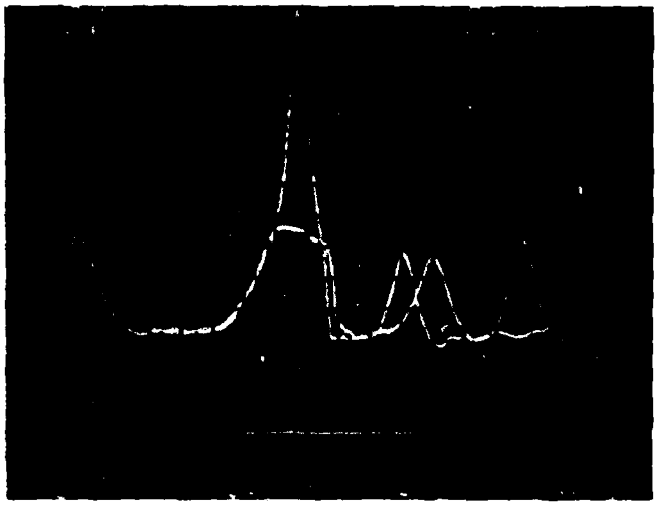
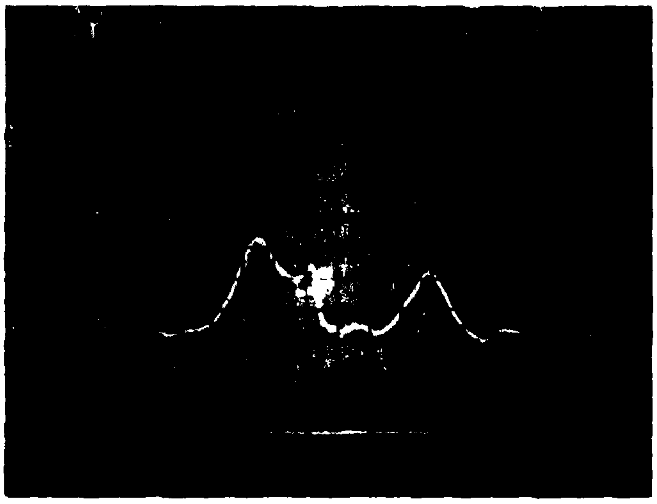



Figure 3.10 Lean misfire limit as a function of ignition timing for neat methanol and solvent ethanol.



PJI



CI

**Figure 3.11** Pressure versus time histories for PJI and CI modes of ignition at the lean misfire point.

chamber are cold, a portion of the methanol vapor can condense onto the walls. As the fuel/air equivalence ratio was reduced, some of the condensed methanol may have evaporated and elevated the equivalence ratio. Since the equivalence ratio was always based on the amount of fuel initially added to the intake manifold, this vaporization of condensed fuel can explain the low value for the lean limit of methanol. Methanol is expected to be readily condensed due to its excessive latent heat of vaporization. The trends observed for methanol are expected to be valid, since the post evaporation of fuel is independent of the mode of ignition.

### 3.2.2 Cold Limit

A limited number of measurements were performed to determine the temperature dependence of the lean limit. The greatest effect of temperature was observed for the neat methanol. As the temperature was reduced from 70° F, the lean limit significantly increased. If the manifold and head were further chilled, severe wetting of the manifold walls, scavenging chamber and cylinder surfaces occurred. This condensation essentially flooded the engine. Once condensation was observed on the engine window, no mixture ignition would occur. Ethanol's lean limit was also sensitive to temperature, however, not to the extent observed for methanol. This may have been due to the additives in the solvent ethanol. The lean limits of gasoline and the alcohol gasoline blends also increased. Fuel additions in excess of the stoichiometric values were required to obtain ignition of the mixture.

These results essentially confirm the literature reported difficulties associated with neat alcohols. The alcohol-

gasoline blends appear to be comparable to gasoline in terms of the cold starting ability. This is possibly due to the extended ignition limits of the alcohol containing fuels.

### 3.3 Engine Performance and Exhaust Emissions

PJI engine performance and exhaust emissions have been studied in an unmodified, commercial, four cycle, single cylinder engine. The primary interest of these measurements was to determine the influence of the ignition system and fuel type on the combustion product distributions produced by the engine. Prior to this study, PJI had been investigated in research type engines and shown to improve the gasoline fueled performance of the engine while having very little influence on the carbon monoxide (CO) and hydrocarbon (HC) exhaust emissions. This section examines the engine performance at approximately 1800 rpm and at an engine load of 1000 watts produced by two types of fuel: 30% ethanol-gasoline and 30% methanol-gasoline. The ignition timing was fixed at 22° before TDC and three types of ignition systems are used, OEM- magneto, high energy convention spark (CI) and segmented gap-PJI. The results of these measurements are compared to similar gasoline data. The engine performance characteristics used in this analysis are brake power ($BP_t$), brake specific fuel consumption (BSFC), brake carbon monoxide (BCO) and brake hydrocarbons (BHC).

#### 3.3.1 Brake Power

Brake power is defined as:

$$BP_t = N * T/5252 \qquad (3)$$

where T is the measure engine test torque and N is the engine speed. Figure 3.12 displays the brake power as a function of the air/fuel ratio (and equivalence ratio, $\phi$) obtained for 30% alcohol blends and gasoline. The results indicate that the 30% ethanol mixture yields slightly improved engine power performance in relation to the other fuels using the OEM ignition system. Methanol containing fuel produced considerably less power with the OEM ignition. The power output of the engine was improved for every fuel using the high energy ignition systems. The operational limits were also extended using the high energy ignition systems. The CI system seemed to yield improved engine power in relation to the segmented gap PJ. These data clearly indicated the advantage of using a high energy ignition system under lean burn combustion conditions. When the engine was operated near the stoichiometric fuel/air value, no real advantage was observed for 30% ethanol while 30% methanol performance was substantially improved.

### 3.3.2 Brake Specific Fuel Consumption

BSFC is defined as:

$$BSFC = F_t/BP_t \quad (4)$$

where $F_t$ is the measured fuel consumption per hour. Figure 3.13 displays the results of this analysis for the three fuels. The high energy ignition systems are found to reduce the engine's fuel consumption in relation to the OEM ignition. The CI ignition system appears to have less improvement for the alcohol containing mixtures. The two high energy systems appear to be indistinguishable in this analysis. The fuel consumption using 30% methanol is greater than 30% ethanol.

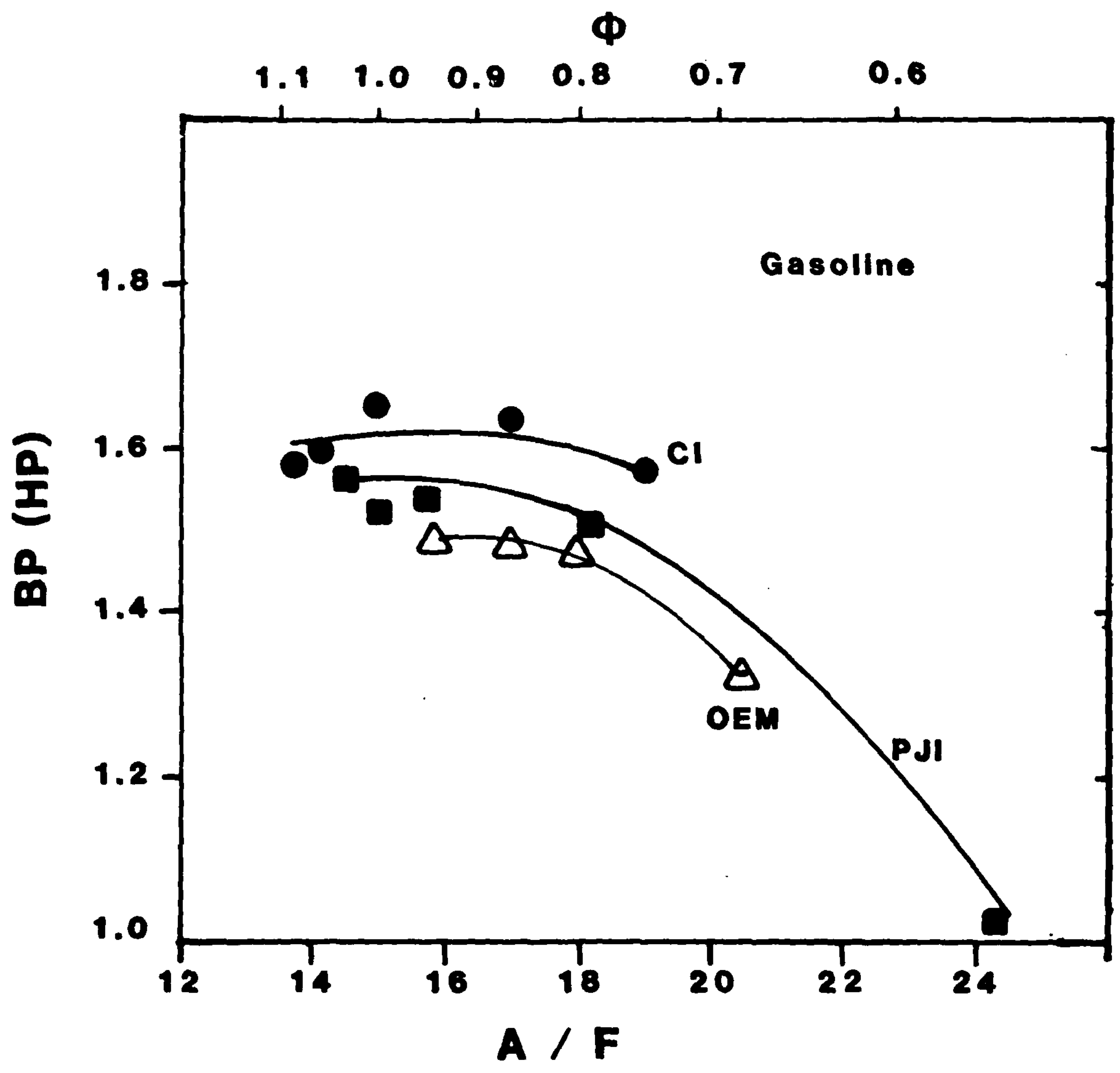



Figure 3.12 Brake power data for the three modes of ignition: OEM, CI and PJI, and three fuels: gasoline, 30% methanol-gasoline and 30% ethanol-gasoline.

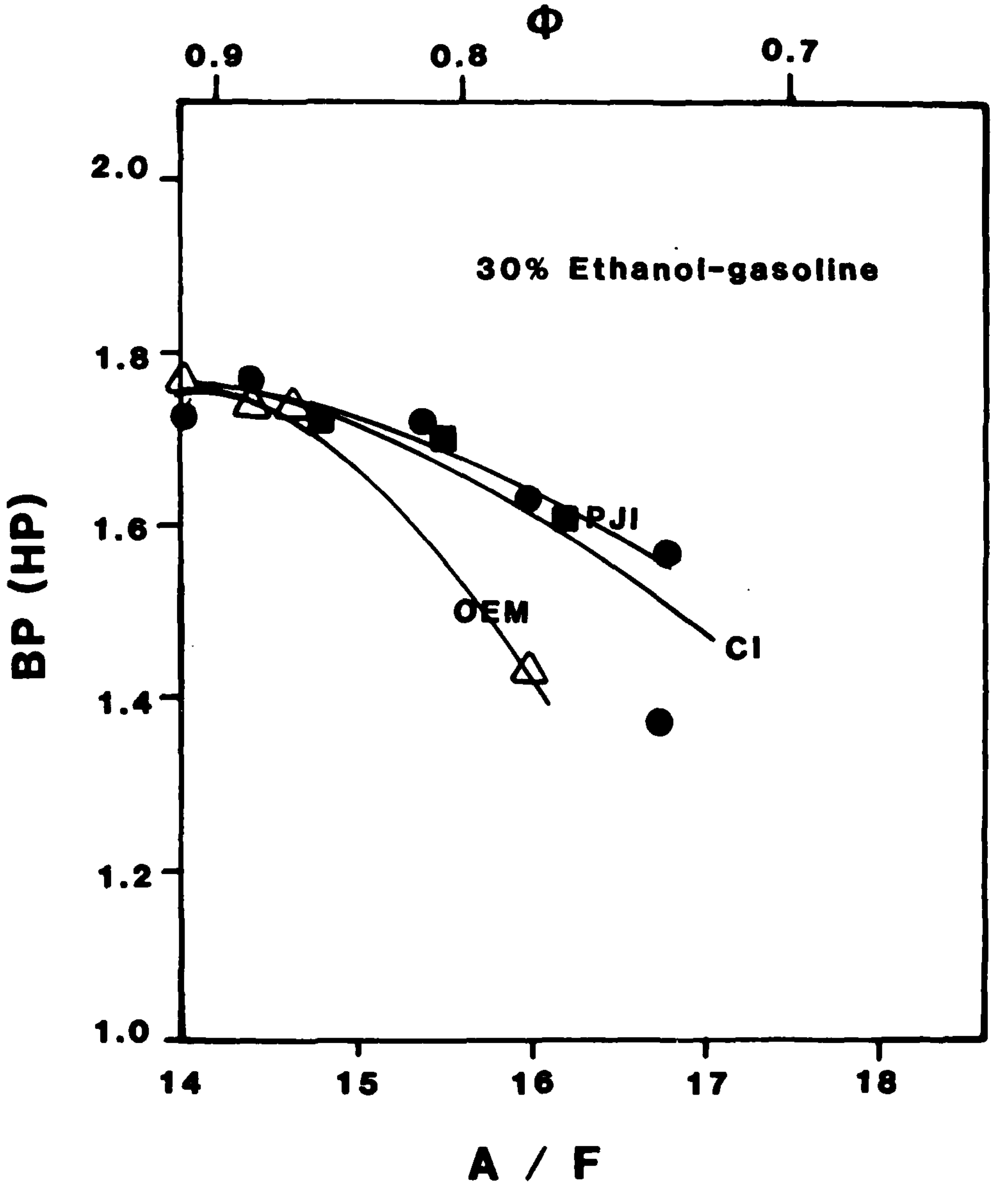



Figure 3.12 Brake power data for the three modes of ignition: OEM, CI and PJI, and three fuels: gasoline, 30% methanol-gasoline and 30% ethanol-gasoline.

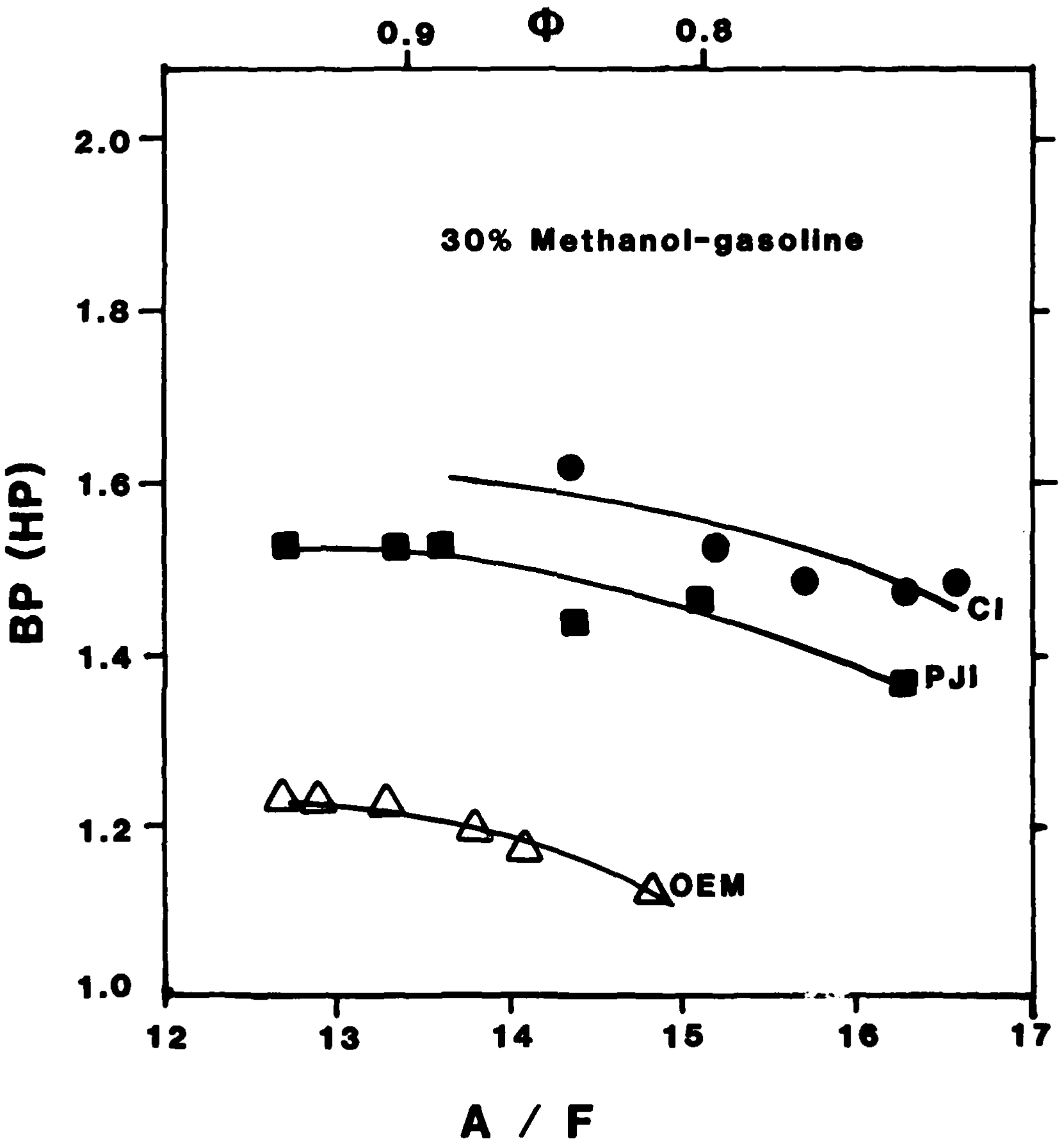



Figure 3.12 Brake power data for the three modes of ignition: OEM, CI and PJI, and three fuels: *gasoline*, 30% methanol-gasoline and 30% ethanol-gasoline.

The improved fuel consumption with ethanol is due to the increased energy content of the ethanol.

### 3.3.3 Brake Specific Emissions

Figures 3.14 and 3.15 display the exhaust emissions determined in this study. Within experimental error, the emission levels appear to be independent of the type of ignition system. This is a confirmation of the results previously reported for PJI. The alcohol containing fuels appear to produce slightly lower amounts of CO in relation to gasoline. This was true for the leaner mixture strengths. However, the 30% methanol blend was found to produce significant aldehyde emissions. These aldehyde emissions were a factor of one-third the hydrocarbon concentrations. In contrast, the aldehyde concentrations for gasoline and the 30% ethanol-gasoline blend were below the detectable limit.

## 4.0 SUMMARY AND CONCLUSIONS

Alcohols, such as ethanol and methanol, are potential substitutes for gasoline during periods of fuel shortages. The pure alcohols have been reported to cause performance and starting problems when used to fuel internal combustion engines. This study characterized how three modes of ignition, OEM-magneto, high energy conventional spark (CI) and plasma jet ignition (PJI) influenced the engine combustion properties of ethanol, methanol and gasoline-alcohol blends. Specific combustion properties examined in these measurements were burning velocity and lean limit. In addition, the engine performance was determined for 30% alcohol-gasoline containing blends. These engine performance measurements determined brake power, brake specific fuel consumption and brake emissions of carbon monoxide and hydrocarbons.

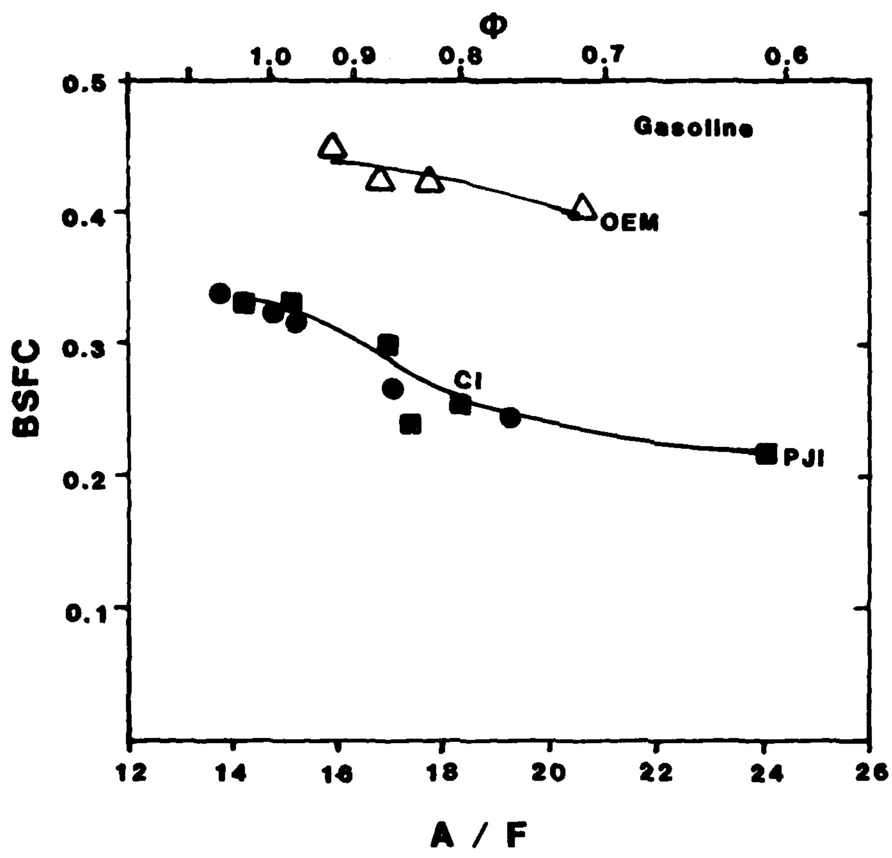



Figure 3.13 Brake specific fuel consumption for the three modes of ignition: OEM, CI and PJI, and three fuels: gasoline, 30% methanol-gasoline and 30% ethanol-gasoline.

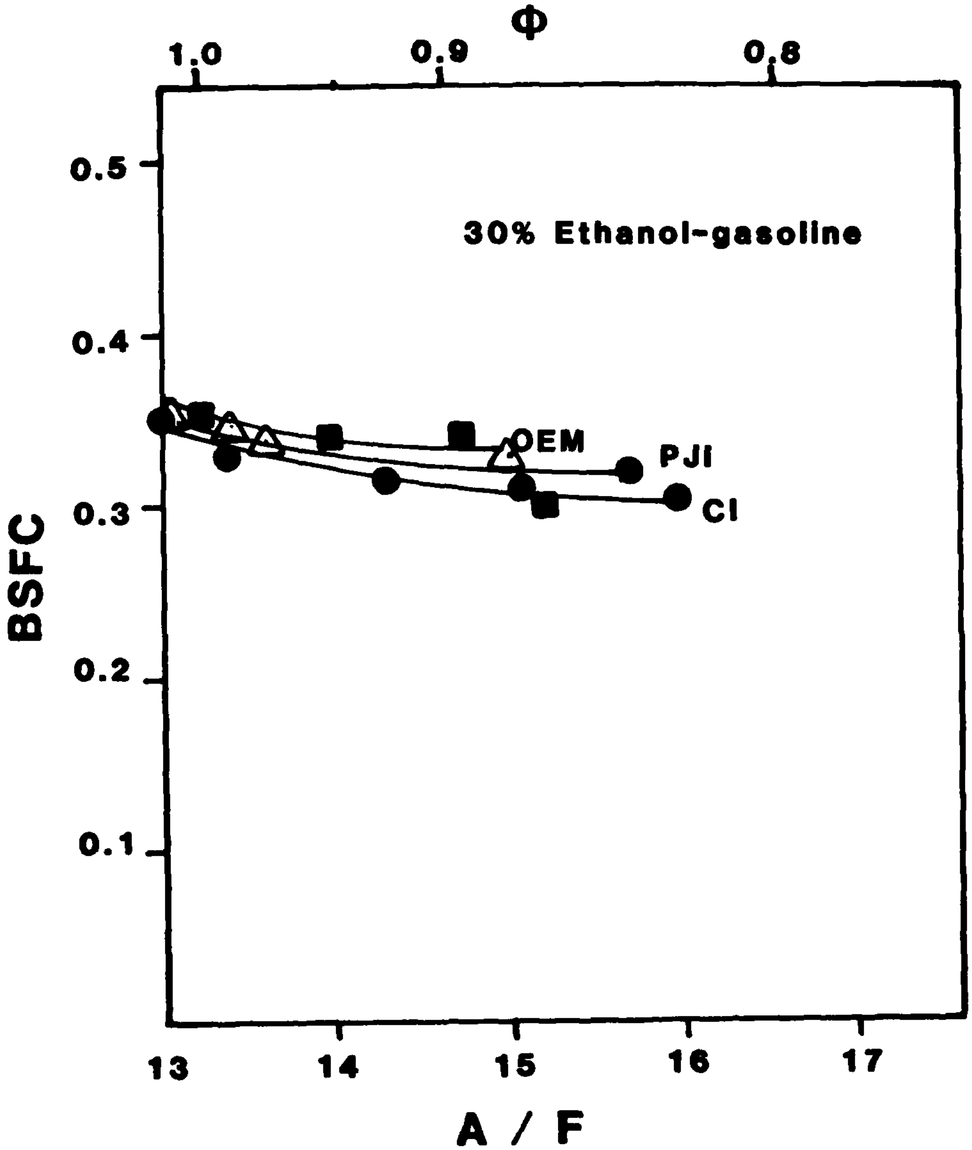



Figure 3.13 Brake specific fuel consumption for the three modes of ignition: OEM, CI and PJI, and three fuels: gasoline, 30% methanol-gasoline and 30% ethanol-gasoline.

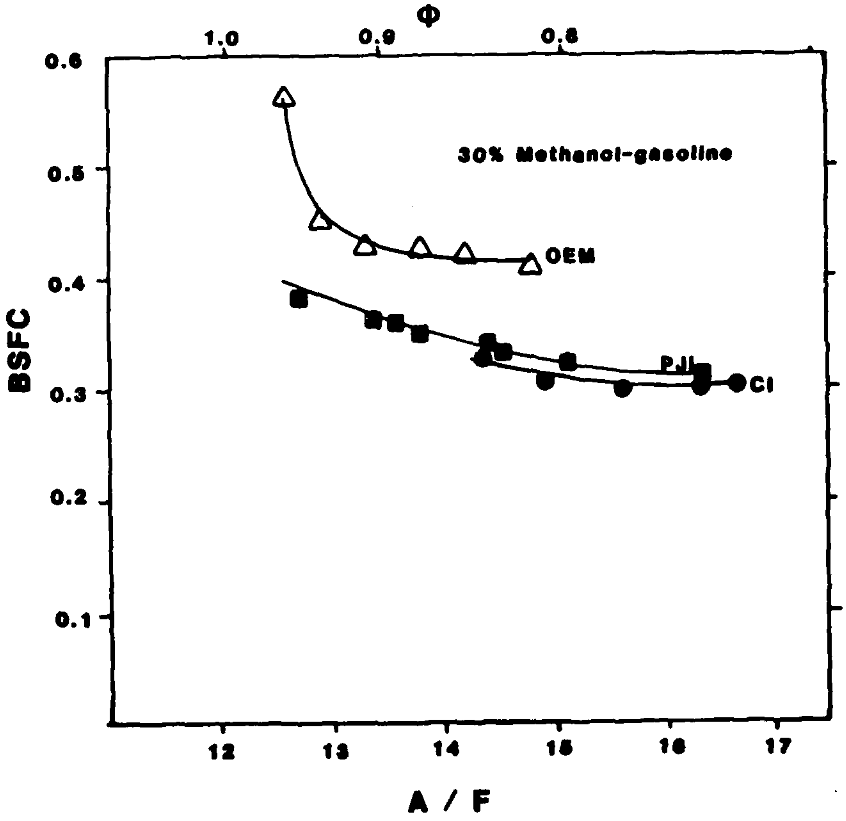



Figure 3.13 Brake specific fuel consumption for the three modes of ignition: OEM, CI and PJI, and three fuels: gasoline, 30% methanol-gasoline and 30% ethanol-gasoline.

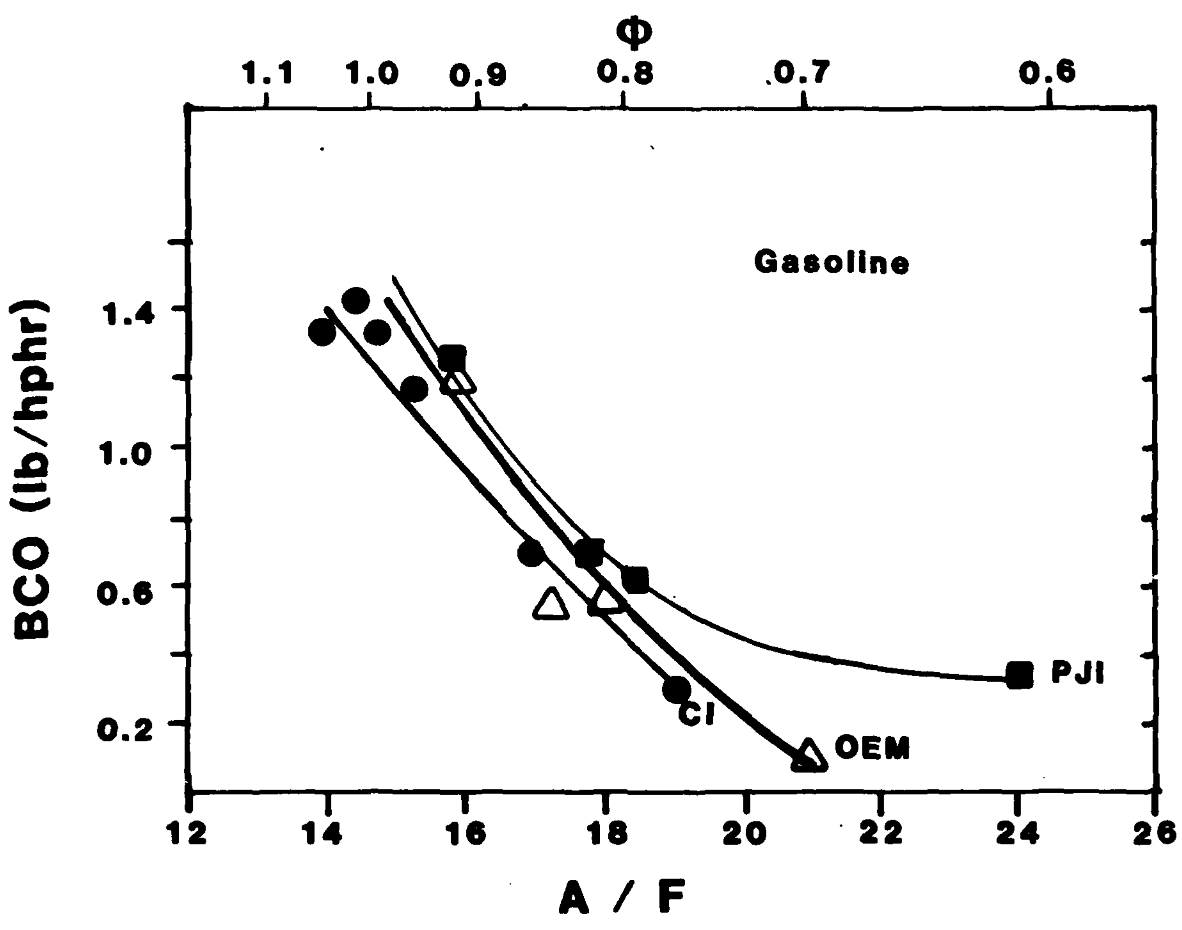



Figure 3.14 Brake specific CO for the three modes of ignition: OEM, CI and PJI, and three fuels: gasoline, 30% methanol-gasoline and 30% ethanol-gasoline.

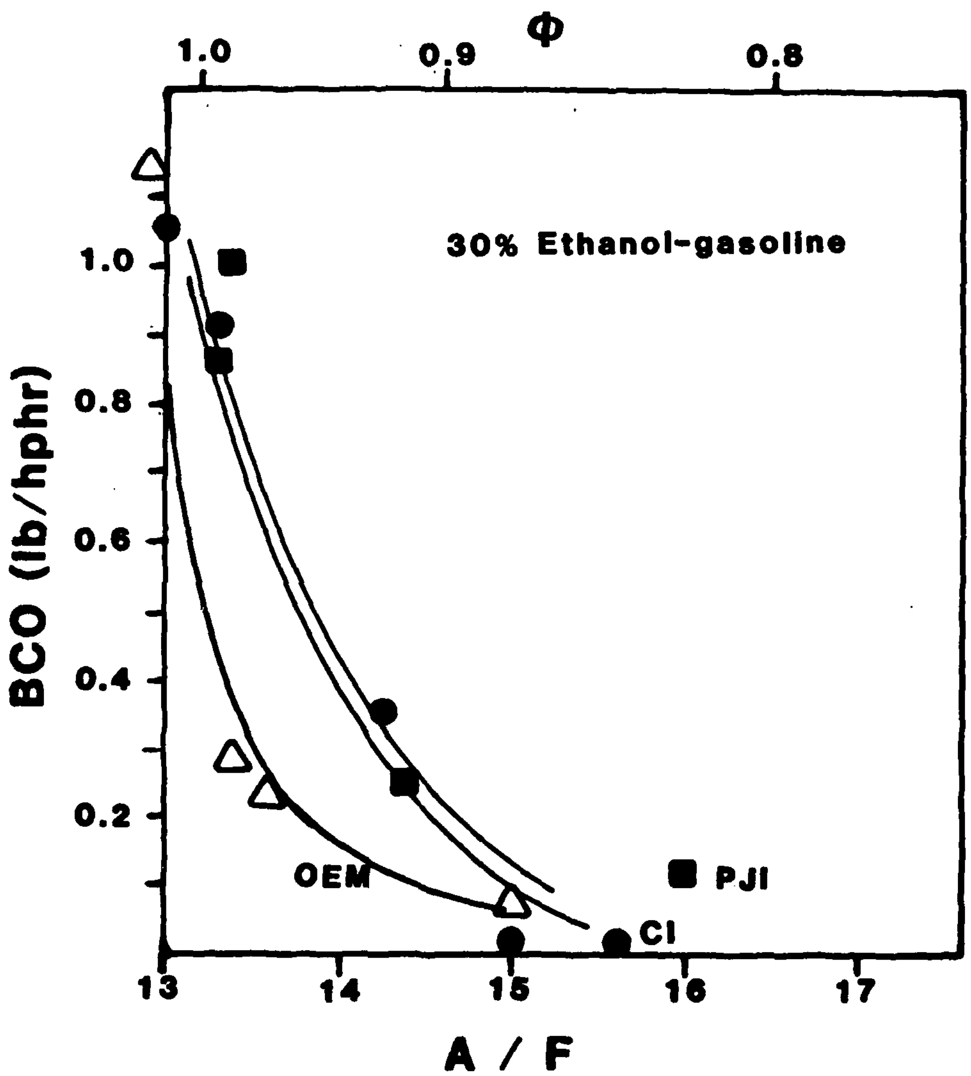



Figure 3.14 Brake specific CO for the three modes of ignition: OEM, CI and PJI, and three fuels: gasoline, 30% methanol-gasoline and 30% ethanol-gasoline.

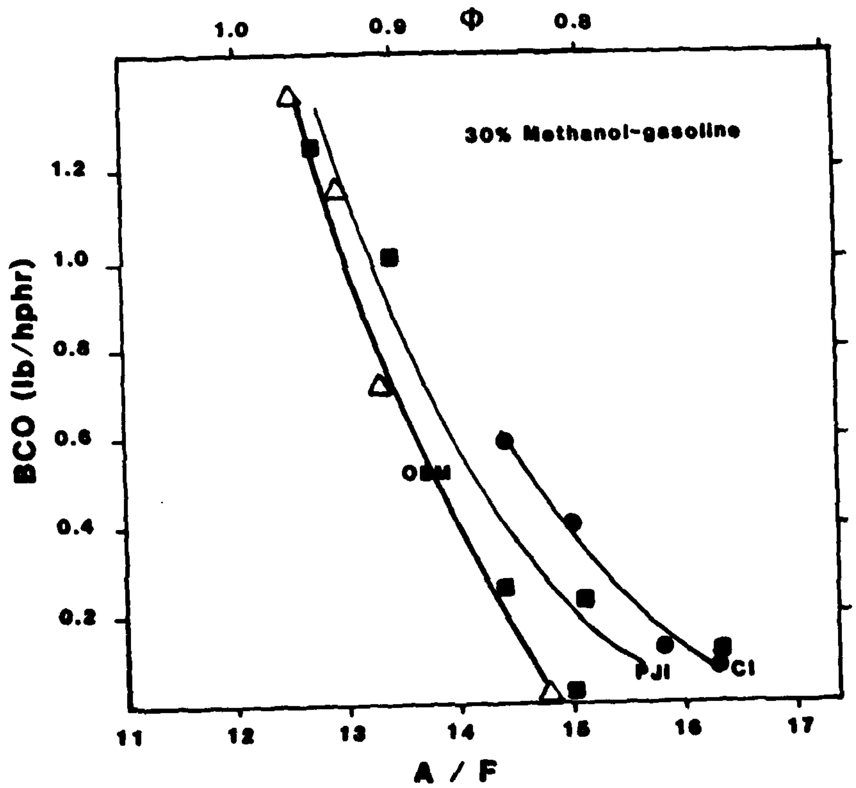



Figure 3.14 Brake specific CO for the three modes of ignition: OEM, CI and PJI, and three fuels: gasoline, 30% methanol-gasoline and 30% ethanol-gasoline.

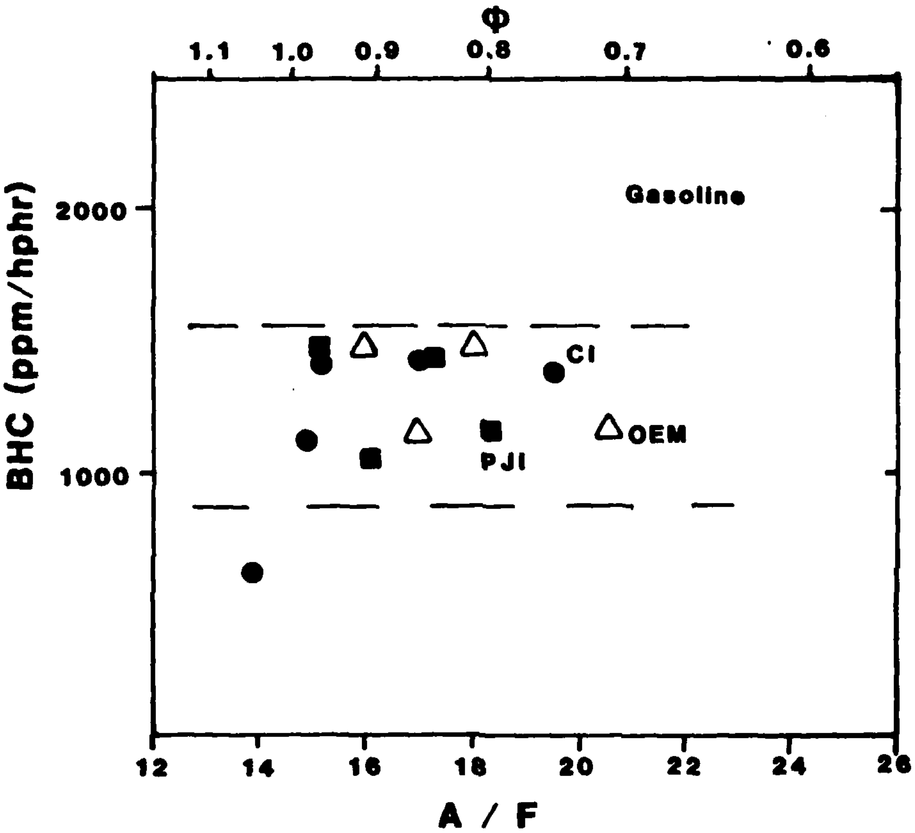



Figure 3.15 Brake specific hydrocarbons for the three modes of ignition: OEM, CI and PJI, and three fuels: gasoline, 30% methanol-gasoline and 30% ethanol-gasoline.

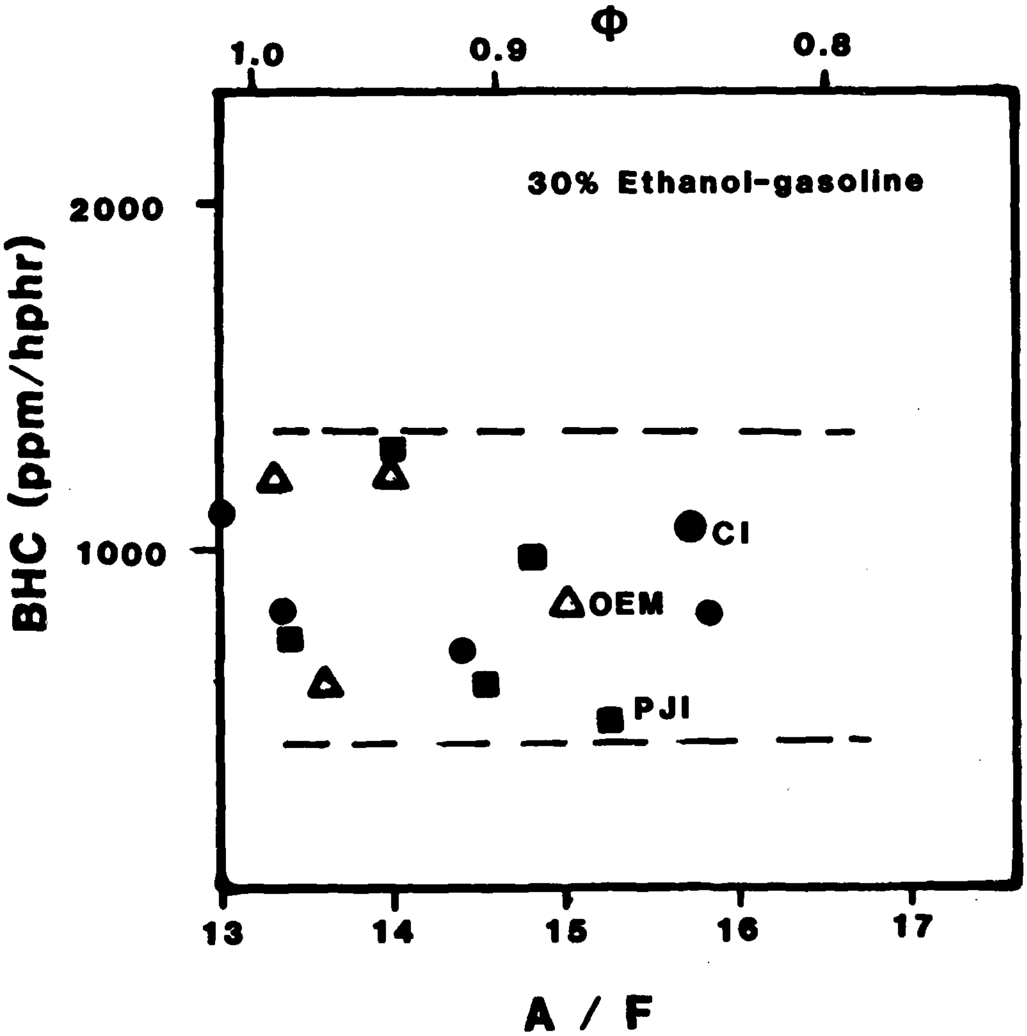



Figure 3.15 Brake specific hydrocarbons for the three modes of ignition: OEM, CI and PJI, and three fuels: gasoline, 30% methanol-gasoline and 30% ethanol-gasoline.

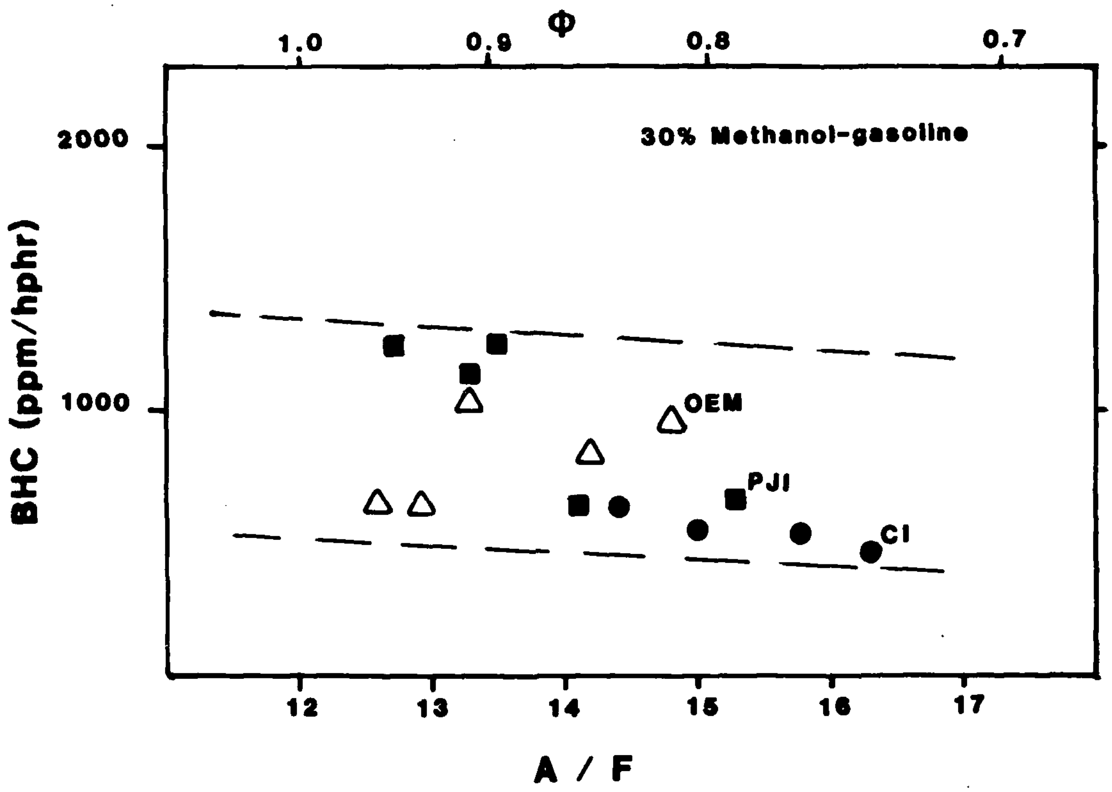



Figure 3.15 Brake specific hydrocarbons for the three modes of ignition: OEM, CI and PJI, and three fuels: gasoline, 30% methanol-gasoline and 30% ethanol-gasoline.

The burning velocity data proved that both the fueled and unfueled PJI enhanced the burning rate of all the fuels in relation to an equal energy CI system. These results further indicated that the fueled PJI produced greater burning velocities than the unfueled PJI geometry. Since reliable engine compatible fueled PJI plugs were not available, these results suggested a developmental effort should be undertaken to construct such a plug geometry.

The burning velocity of all alcohol containing mixtures exhibited higher burning velocities in relation to gasoline at one atmosphere initial pressure. However, measurements in the optical engine found that gasoline burned at a higher rate. In addition, mixtures containing ethanol were found to have higher burning velocities in relation to analogous methanol mixtures. The optical engine burning velocity data suggested that the ignition timing of the engine should be retarded from the gasoline value when alcohol blends were used. This was suggested by the sizable ignition delays determined for the alcohol blends.

The lean limit of each fuel was determined for warm and cold starting conditions. Two types of ignition, CI and unfueled PJI, were tested. The unfueled PJI was found to extend the lean limit of all fuels by approximately 10% in relation to the lean limit determined for the CI mode of ignition.

The warm lean limit measurements found that gasoline had the highest lean limits in relation to all other fuels tested. Neat methanol and solvent ethanol had the lowest limits. The addition of alcohol to gasoline was found to lower the ignition limit. The cold limit study found that

methanol vapors readily condensed in the engine and resulted in flooding. The engine charge was not ignitable under these conditions. Large amounts of ethanol were required to obtain ignition under cold conditions. The additives in the ethanol were suggested to be responsible for the improved ignition in relation to methanol. Gasoline and the alcohol-gasoline blends were ignitable under cold conditions. However, equivalence ratios in excess of the stoichiometric value were required to obtain reproducible ignition. These measurements under cold conditions confirmed the cold start difficulties reported for methanol.

The engine performance testing found that the CI and unfueled PJI resulted in improved brake power, brake specific fuel consumption and wider air/fuel ratio operational limits. All ignition systems produced comparable emission levels.

The 30% ethanol-gasoline blend was found to slightly improve the brake power and widen the operational fuel/air limits of the engine in relation to neat gasoline. The 30% methanol-gasoline blend was found to have reduced power and elevated aldehyde exhaust emissions in relation to both neat gasoline and 30% ethanol-gasoline. Based on the results of these measurements, the addition of amounts of alcohol up to 30% do not significantly affect the engine performance, while the addition of the alcohol extends the air/fuel ratios over which the engine will operate. The brake specific fuel consumption results suggested that the 30% blends do not produce a dramatic increase in the engine fuel consumption of the engine.

These results further suggest that ethanol-gasoline blends will produce comparable engine performance to gas-

oline, will have an extended lean limit and will not be significantly affected by cold temperatures (<0°C). Ethanol is thus suggested to be the better gasoline substitute in relation to methanol. Of course, sufficient quantities of ethanol must be available to make this practical.

## 5.0 RECOMMENDATIONS

The findings of this alternate fuel ignition and performance study suggest that a rugged, longlived, fueled PJI plug should be developed and tested. In addition, a more extensive evaluation of both high energy ignition systems, PJI and CI, should be performed in multi-cylinder engines.

This study recommends that the fuel delivery system of most small gasoline powered engines should be modified to improve their alternate fuel compatibility, especially for the alcohol fuels. Two such modifications are recommended, an intake air heater and fuel injection. The low volatility and high latent heat of vaporization of the alcohols will cause condensation of alcohol in the engine and poor starting ability of the engine. These problems will be especially severe for methanol. A heater placed in the air intake to warm the incoming air will reduce the severity of these low temperature problems. Since the stoichiometric air/fuel ratios for ethanol and methanol, 9 and 6.4 respectively, are considerably different from gasoline (14.6), considerable recalibration of the standard small engine carburetor will be required to operate these engines on alcohols. The electronic control of fuel injection systems will greatly simplify the conversion of the engine for different fuel types. However, fuel injection represents additional engine

costs, component count and complexity of repair.

Geo-Centers, Inc. recommends a further extension of the Phase I study. A two year program is proposed to investigate the potential of high energy ignition to improve the performance of small and medium sized gasoline engines. Specific fuels to be studied include gasoline, kerosene, alcohols, alcohol blends, and synthetic fuels.

Specifically, Geo-Centers, Inc. proposes five tasks to accomplish these objectives. The following tasks are to be accomplished over a two year period:

1. Construct rugged, long-lived fueled and unfueled, segmented gap plasma jet plugs. These plugs will be manufactured by a commercial spark plug producer.

2. Perform an extensive series of single cylinder engine performance measurements to evaluate the influence of the anticipated enhanced combustion rate on engine power, response to load and fuel consumption.

3. Evaluate the multi-cylinder performance for gasoline, kerosene, alcohols and alcohol-gasoline blends using the high energy and OEM ignition systems tested in Phase I. The engine types to be tested will be specified by the COR and provided by the MERADCOM electrical power laboratory (GFE).

4. Repeat the testing defined in Task 3, and evaluate the engine starting ability in the MERADCOM environmentally controlled test cell.

5. Prepare written and oral reports to the COR and a final report summarizing the progress and results of the program, as required by the COR.

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